REDUCING EMISSIONS FROM A NATURAL GAS POWERED WANKEL ENGINE

A THESIS

Presented to

The Faculty of the Division of Graduate

Studies and Research

By

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In Partial Fulfillment

of the Requirements for the Degree

Master of Science in Mechanical Engineering

Georgia Institute of Technology

December, 1972

REDUCING EMISSIONS FROM A NATURAL GAS

POWERED WANKEL ENGINE

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Date approved by Chairman: March 9, 1973

ACKNOWLEDGMENTS

I owe a great deal to many people for their help toward the completion of this thesis.

I would like to give special thanks to my advisor, Dr. Samuel V. Shelton, whose guidance was of utmost importance to me.

I also wish to thank Dr. Pandeli Durbetaki and Dr. Michael Matteson for serving on my reading committee.

Mr. Joe Doyal, Mr. Lou Cavalli, and Mr. Donald Cabe, all of whom are School of Mechanical Engineering technicians, deserve thanks for their constant help and good humor.

I am indebted to Mr. John Sholar and the Atlanta Gas Light Company for providing the engine and other equipment used in this work.

Thanks go also to two fellow students, Mr. Marion Pacetti and Mr. William Clark, for their help in setting up equipment and running tests on numerous occasions.

Finally, I would like to dedicate this thesis to my wife, Susan. Special thanks are due her for the sacrifices she has made for my continued education, and for the help and understanding that she has given me during my time at Georgia Tech.

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SUMMARY

Automobile engines have been proven to be a major source of air pollutants, primarily hydrocarbons, carbon monoxide, and oxides of nitrogen. As a result, Federal law has required that by 1976 each of these automobile engine exhaust pollutants must be reduced by 90 per cent from their corresponding levels in 1970 model cars.

The Wankel engine has jumped into prominence in recent years as a possible solution to the problem of meeting the 1976 Federal emissions standards. Toyo Kogyo has claimed that they can meet the standards at the present time with the Wankel, but definite proof of their claims has not yet been verified.

Natural gas has been discussed in recent years as an alternative fuel for the internal combustion engine in order to help it meet the 1976 emissions standards. Evidence has shown that a natural gas powered engine results in less pollutants being emitted than in a gasoline powered engine, provided neither engine possesses external emission control devices.

The objective of this study was to determine how to minimize exhaust emissions from a natural gas powered Wankel which utilized a thermal reactor and a catalytic converter. These minimized exhaust emission levels were to then be evaluated by comparing them to the levels of pollutants emitted from the same engine while being run on propane and gasoline.

In minimizing engine exhaust pollutant levels, the following parameters were investigated: engine rpm, engine load, air-fuel ratio, and the amount of air injection into the exhaust system.

It was found that exhaust emissions were minimized on a total basis if the air-fuel ratio was between 19 and 20, the emgine rpm high, the engine load moderate, and the air injection into the exhaust system at its maximum value. A major setback, however, to the use of natural gas for the purpose of reducing hydrocarbon exhaust emissions was discovered. This was the fact that present exhaust emission control devices were relatively inefficient in reducing exhaust hydrocarbon emissions which resulted from the use of natural gas as a fuel. Average HC reductions of only 7 per cent in the thermal reactor and 20 per cent in the catalytic converter were found.

On a total exhaust emission basis, natural gas appeared to have less potential than propane for becoming the fuel that would operate the cleaner Wankel.

CHAPTER I

INTRODUCTION

A History of Automotive Pollution and Emission Standards

The role of the automobile in the nation's pollution problem first became apparent as a result of studies of the air pollution problem in Los Angeles, California. It was found in the early 1940's in the Los Angeles area that air pollution was causing plant damage, eye irritation, cracking of stressed rubber, and visibility reduction. Initial efforts to solve the problem were aimed at reducing particulate emissions from industries, and the particulates were reduced by two-thirds. The smog and its related symptoms, however, remained.

A research program was undertaken to find the cause or causes of this new type of air pollution. It was found that this smog was very oxidizing in nature because of the high ozone content, which was contrary to the reducing atmosphere normally connected with the sulfur dioxide emissions from powerplants and home furnaces. Since no significant amounts of ozone were released into the air in the Los Angeles area, some form of atmospheric reaction was suspected. In 1952 Professor A. J. Haagen-Smit at the California Institute of Technology demonstrated that hydrocarbon compounds reacted in the presence of sunlight and NO_2 to form many oxidation products that could account for

the smog (1).*

The automobile was studied extensively thereafter and was found to be a substantial contributor to the total hydrocarbon (HC) emissions, carbon monoxide (CO) emissions, and nitric oxide (NO_x) emissions. By 1958, with smog becoming worse every year, it was apparent that some type of automobile HC emission control would be required for Los Angeles. Subsequently, the state of California in 1959 initiated the control of motor vehicle emissions with the adoption of standards to control emissions resulting from crankcase blowby (2).

The 1965 Amendments to the Federal Clean Air Act gave the Secretary of the Department of Health, Education, and Welfare, the authority to control emissions from motor vehicles. The initial Federal motor vehicle emission standards were adopted in March, 1966, to become effective with the 1968 models. The standards and procedures were similar to those previously set by California. Then in June, 1968, Federal standards were revised and required more stringent HC and CO control, as well as tight controls on evaporative emissions from fuel tanks and carburetors. The fuel evaporative emission standards became fully effective on 1971 model vehicles. The other standards applied to 1970 model year vehicles and engines. Thus with the arrival of 1970 models, hydrocarbon emissions had been reduced by almost three-quarters and CO emissions by about two-thirds from uncontrolled engines (2).

Exhaust emissions were measured initially on a concentration

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*Numbers in parentheses refer to references in the Bibliography section.

basis. The vehicle was placed on a chassis dynamometer and was operated over a predetermined driving cycle. The driving cycle used in California to certify 1966 and newer vehicles and by the Federal government to certify 1968-71 vehicles has commonly been referred to as either the California cycle or Federal Test Procedure (FTP) (1).

Air pollution, however, depends upon the concentration of pollutant emitted and the volume, and the 1970 and 1971 Federal standards were therefore based on a calculated mass emission rather than just concentration. Average vehicle emission concentrations were obtained with the FTP driving cycle and were multiplied by the total cycle calculated exhaust volume. This mass emission result was not a true mass emission measurement since actual exhaust volumes were not measured during the tests. Instead a multiplier was used which was a function of the vehicle weight only. A true mass emission measurement procedure has been developed for use on 1972 and newer cars. This procedure employs variable dilution sampling, or constant volume sampling (CVS), and it employs a new driving cycle. For 1972-74 cars, a single gas sampling bag is used and the cycle is called CVS-1 (1). For 1975 and later model cars, the CVS-3 test will be used. This test is basically an averaging of both a hot and cold engine test cycle.

Table 1 gives a summary of Federal emission standards for automobiles. It should be noted that control began in 1971 to reduce NO_x for smog and health effects. Particulate emission control of 0.1 gm/mile has been proposed for 1975 mainly for health reasons. The 1976 standards represent a 90 per cent reduction in HC, CO, and NO_x from the

standards for 1970 automobiles.

The Department of Health, Education, and Welfare has estimated that on a national basis motor vehicles account for approximately 39 per cent by weight of the air pollution problem (3). H.E.W. also estimated that automobiles contribute about 48 per cent of all the CO, 4 per cent of the sulfur dioxide (SO_2), 32 per cent of the NO_x, 59 per cent of the HC, and 8 per cent of the particulate matter in the air (1). The particulates are mostly lead from engine exhaust, asbestos from brake linings, and rubber from tires.

Table 1. Past, Present, and Future Federal Automotive Exhaust Emission Standards

Year	HC	CO	NO_x
FTP Cycle			
1968	275 ppm	1.5 %	-
	(3.2 gm/mi)	(33 gm/mi)	-
1970	2.2 gm/mi	23 gm/mi	-
	(180 ppm)	(1.0 %)	-
1971	2.2 gm/mi	23 gm/mi	4.0 gm/mi*
CVS-1 Cycle			
1971	4.6 gm/mi	47 gm/mi	-
1972	3.4 gm/mi	39 gm/mi	4.0 gm/mi
1973-74	3.4 gm/mi	39 gm/mi	3.0 gm/mi
CVS-3 Cycle			
1975	0.41 gm/mi	3.4 gm/mi	3.0 gm/mi
1976	0.41 gm/mi	3.4 gm/mi	0.4 gm/mi

Evaporative losses: 6.0 gm per test in 1971 nationwide, and 2.0 gm per test in 1972 nationwide.

*California requirement only.

The fact that the motor vehicle accounts for 39 per cent by weight of the nation's air pollution problem may be misleading in urban areas. For instance it has been estimated that 80 to 90 per cent of the Los Angeles pollutants and around 80 per cent by weight of the Atlanta pollutants are contributed by motor vehicles.

Smog is not the only detrimental effect resulting from automobile pollutants. Plants are damaged to various extents by hydrocarbons, nitric oxides, and photochemical oxidants. Sufficiently high concentrations of NO_x and photochemical oxidants lead to respiratory disorders in humans, especially in the young and the aged. In addition, CO is well known for its poisonous effects on man at high concentrations. It is absorbed by the hemoglobin in the blood much easier than is oxygen and, thus, decreases the oxygen capacity of the blood. CO has been associated with blurry vision, dizziness, and reflex impairment at moderate concentrations (2).

The automobile has definitely been found to be detrimental to society because of its pollution capabilities. The automobile industry, recognizing this fact and under pressure now from the Federal government, has been taking steps to correct this problem. These steps will be dealt with in the following section.

Present and Future Means of Automobile Emission Control

Of the total hydrocarbon emissions from an uncontrolled automobile, 20-25 per cent arises from crankcase blowby, 60 per cent from the exhaust and the remainder from evaporative losses from both the carburetor and the fuel tank (1). Formerly the blowby gases from the

cylinders into the crankcase were vented directly to the atmosphere. This crankcase blowby problem has now been eliminated with the addition of the positive crankcase ventilation (PCV) system which puts the pollutants back into the intake manifold to be recycled. The breather cap is opened to the clean side of the air cleaner to allow air flow and to increase purge rates. Control of recirculated blowby volume is achieved with a vacuum sensitive valve (PCV valve) with a variable orifice. This PCV system has completely eliminated the crankcase blowby pollution problem and no future control is required.

In addition, evaporative control systems have been designed to virtually eliminate HC vapors emitted by the carburetor and fuel tank during running and hot soak. When the engine is running, carburetor losses are vented into the intake system and fuel tank vapors are inducted and burned in the engine. During hot soaks, fuel tank vapors are routed to a storage device and carburetor vapors are either vented to the storage system or retained internally in the carburetor or induction system volume. Upon restart, filtered air is drawn through the stored volumes and is metered to the intake system. This air purges the storage device. The polluting vapors are stored in either a cannister containing activated charcoal or in the engine crankcase.

Engine exhaust emissions, primarily HC, CO, and NO_x, constitute the major automobile pollution problem and must be controlled. Engine speed, air-fuel ratio, mixture distribution, and spark timing all have significant effects on cylinder combustion conditions, and, thus, determine these pollutant concentrations. HC and CO are dependent on the

amount of incomplete combustion of the air-fuel mixture, while NO_x concentration is dependent on peak combustion temperature and available oxygen. Therefore, attempts to lower HC and CO emissions will almost always result in a higher emission of NO_x.

Changes in engine operating conditions and some engine design modifications have been carried out to date by the various automobile manufacturers and have resulted in a reduction of exhaust emissions. Carburetor adjustments have been made to give leaner air-fuel mixtures and better mixture control. In addition, intake air heating, increased idle speed, retarded spark timing, improved cylinder head design, and reduced engine compression ratios have improved vehicle emissions up to this point.

Exhaust gas recycle (EGR) has already been installed on many automobiles today and will be installed on the others in the future to meet the 1976 NO_x standards. EGR is the recycling of 10 to 20 per cent of the exhaust gases to the intake manifold to dilute the new air-fuel mixture that enters into the combustion chambers with an almost inert gas. This recycled exhaust takes up some of the room in the combustion chambers and lowers peak combustion temperatures, thus reducing the formation of NO_x. The amount of exhaust gas recycled is matched to engine conditions by a valve activated by manifold vacuum or throttle position.

Thermal reactors, already standard on Mazda rotary engines, are being considered for future HC and CO reduction. These reactors, which bolt directly onto the exhaust ports, promote rapid mixing of the hot exhaust gas with secondary air supplied by an air pump and retain the

gases at a high temperature for enough time to burn up most of the exhaust HC and CO. The air-fuel mixture in these engines with reactors is enriched so that the chemical energy released inside the reactor in the burn-up process holds the high temperature.

Catalytic converters have also been proposed for future automobiles to reduce HC and CO emissions. By means of catalysts the HC and CO in the exhaust gas are oxidized at lower temperatures than those in a thermal reactor. Enriching the air-fuel mixture is not necessary, either, since conversion efficiencies of a catalytic converter are usually independent of intake compositions (4). Key features of most converters include the use of low mass, high surface area, reinforced, thermally stable structures designed to direct the exhaust flow through the catalyst bed (5). These proposed converters would require an unleaded fuel.

All present and proposed exhaust emission control techniques have a large impact on engine operation and performance. If each component in these systems is not well integrated into the whole operation the system performance will not give nearly the desired results. Various engine modes in everyday driving necessitate the fact that the air-fuel ratio, air volume intake, and spark timing all must be varied as engine loads and speeds change. The car must, in addition, start and run well over a wide range of temperatures, altitudes, and humidities. Figure 1 indicates the estimated cost per car of achieving the control standards of 1975. One can readily see that the job of drastically lowering automobile emissions and keeping the engines running

Table 2. Typical Pattern of Installation of Automotive Emission Hardware

Model Year Added	Item
1966	PCV valve
1968	Fuel-evaporation-control system
1970	a. Retarded ignition timing b. Decreased compression ratio c. Change of fuel/air ratio d. Transmission control system
1972	a. Anti-dieseling solenoid valve b. Thermostatic air valve c. Choke-heat by-pass
1973*	a. Exhaust-gas recirculation b. Air-injection reactor c. Induction hardened valve seats d. Spark-advance control e. Air pump
1974*	Precision cams, bores, and pistons
1975*	a. Proportional exhaust-gas recirculation b. Carburetor with altitude compensation c. Advanced air-injection control d. Air/fuel preheater e. Electric choke f. Electronic distributor (pointless) g. Improved timing control h. Catalytic (oxidizing) converter i. Catalyst pellet charge (NO reducing) j. Cooling-system changes k. Improved underhood materials l. Body revisions

*Proposed.

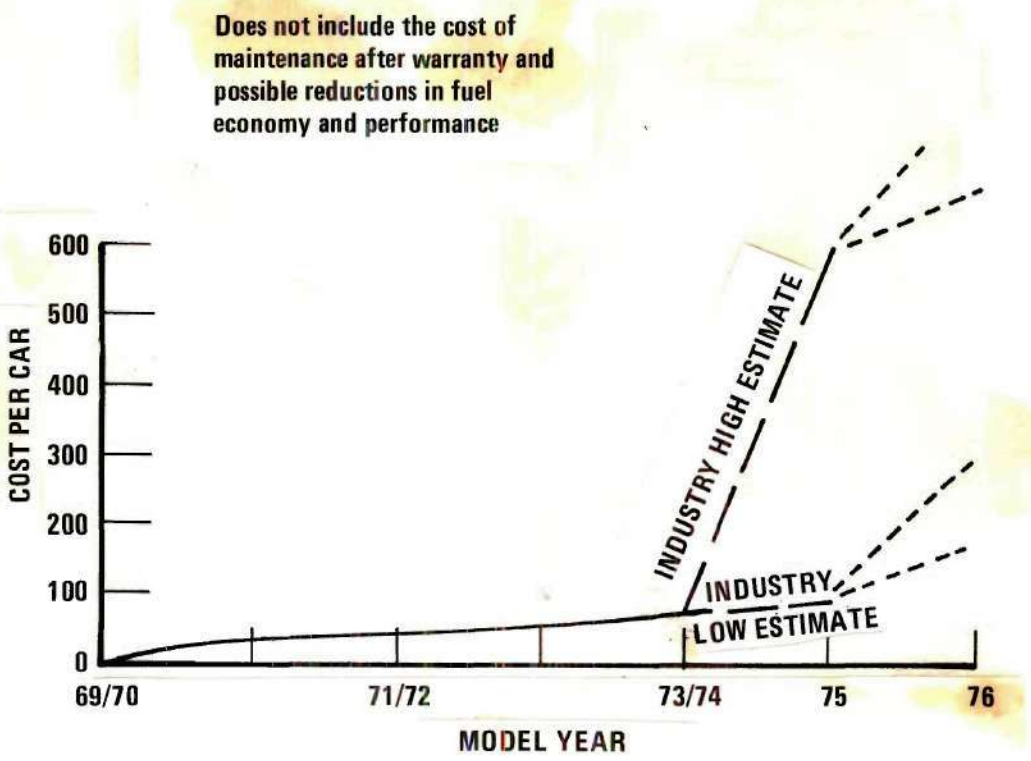

Figure 1. Initial Cost per Car of Emission Control Systems

% REDUCTIONS OF POLLUTANTS REQUIRED BY FEDERAL STANDARDS COMPARED TO UNCONTROLLED VEHICLES

	70/71	72/74	75	76
HC	73	80	98	98
CO	62	69	97	97
NO_x	0	25	25	90

satisfactorily is indeed a complex and expensive assignment.

The Wankel Engine

The Wankel engine was invented in 1954 by Felix Wankel, a German inventor. It was basically a rotary engine on which the Otto, or 4-stroke, cycle principle was applied. In the early stages of development, a Dr. Froëde of NSU simplified Wankel's design by performing a kinematic inversion. He made the inner rotor orbit while it rotated and thus permitted the outer body to be kept stationary. In Wankel's original design, both the rotor and the outer body rotated. Froëde's work also permitted the attachment of the intake and exhaust systems, the ignition, and the coolant connections to a stationary member (6).

In the early stages of development most engineers expressed doubt that the Wankel would ever become a practical reality because of several problems inherent in the design. In recent years, however, the problems have been resolved sufficiently for versions of the engine to be in moderate scale production in both Germany and Japan. Japanese Mazdas with the rotary engine have been selling quite well in the U.S. for the past two years. With recent trends in the U.S. towards the use of low-lead gasoline, sub-compact cars, and lower levels of allowable automobile emissions, the Wankel engine fits in well with its characteristic low NO_x emissions and its ability to operate satisfactorily on either leaded or unleaded gasoline of about 90 octane number or less (6).

As mentioned earlier the Wankel engine operates on the Otto cycle and has the same processes as the conventional reciprocating internal combustion engine. The Wankel, however, is extremely simple

in design. Fundamentally there are only two moving parts, the rotor and the eccentric, or output, shaft. The rotor revolves directly on the eccentric shaft which abolishes the need for connecting rods. Intake and exhaust gases pass through ports so the valves and valve operating mechanism of the reciprocating engine are not required. Figure 2 shows an exploded view of the principal engine parts. One important fact is that the output torque is transmitted to the shaft through the eccentric. The internal and external gears shown are timing gears designed to maintain the phase relationship between the rotor and eccentric shaft rotation. The smaller external gear (coaxial with the eccentric shaft) is actually fixed to one side housing.

Since the rotor is three-sided, there are actually three separate cycles occurring simultaneously. Figure 3 shows the processes which occur in sequence in the Wankel. Figure 3a shows the rotor position during the intake portion of one side of the rotor, Figure 3b during the compression portion, Figure 3c during the ignition portion, Figure 3d during the expansion cycle, and Figure 3e during the exhaust process. Therefore, there are three power impulses for each rotor revolution. However, since the eccentric, or output, shaft rotates at three times the speed of the rotor, there is only one power impulse for each revolution of the output shaft of a single rotor engine.

Even though the simplicity of the Wankel engine is attractive, many problems have arisen during the engine's development. The sealing of the combustion chamber is definitely more difficult than in the reciprocating engine. The rotary engine, which requires gas seals at

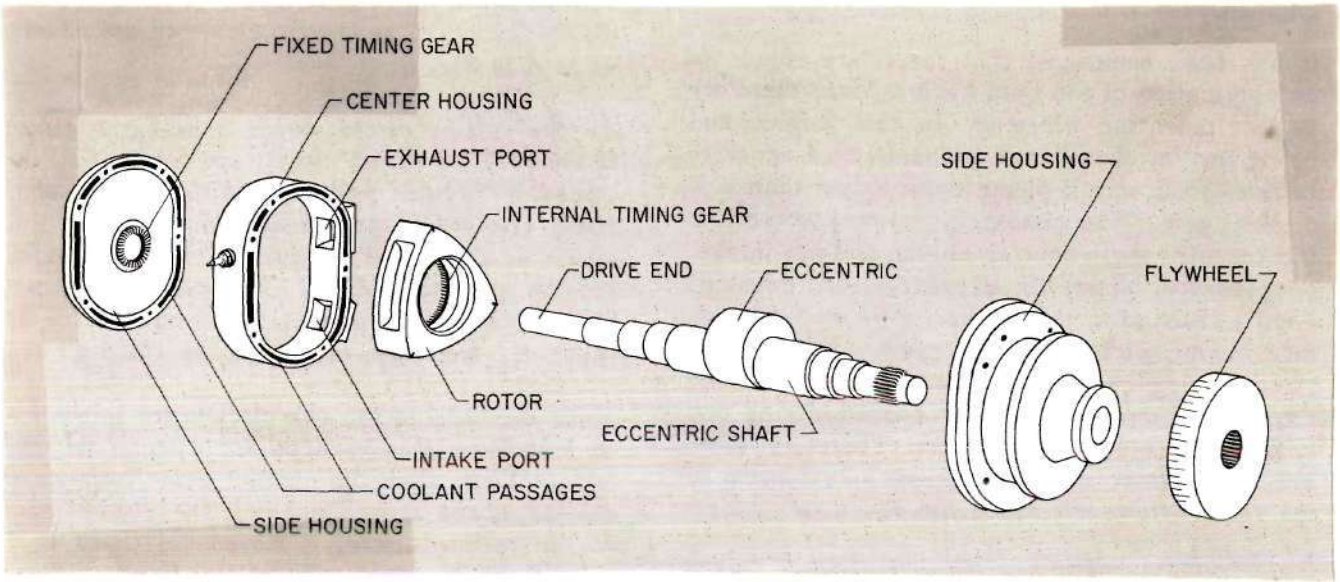

Figure 2. Parts of Rotary Engine

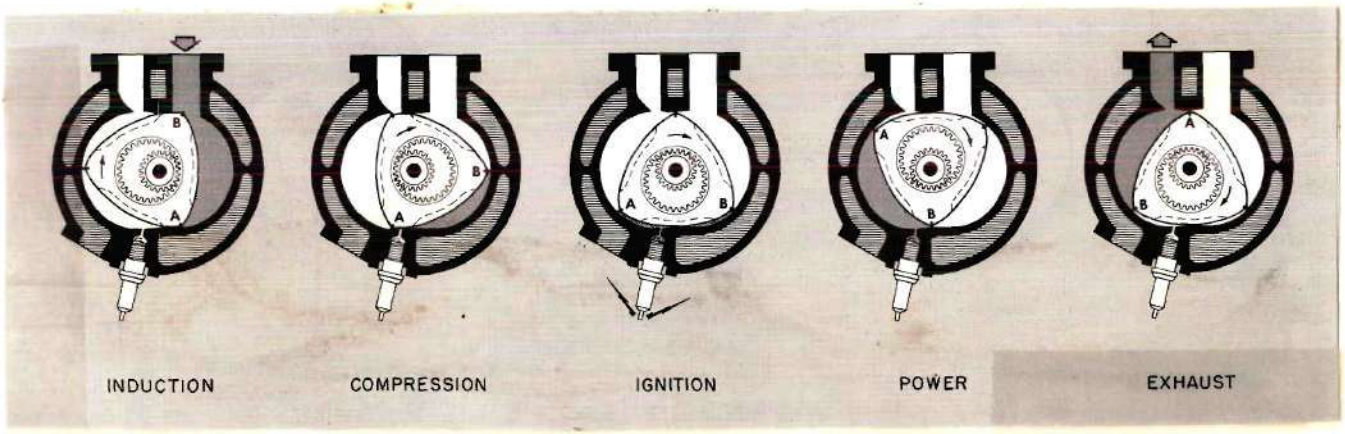

Figure 3. Rotary Engine Cycle

the rotor apexes and around the perimeters of the rotor sides, also requires oil seals on the rotor sides around the bore to minimize oil leakage into the combustion zone (6). Oil, which is used for engine lubrication, of course, is also used to the rotor.

The rotary engine's combustion chamber possesses a higher surface area to chamber volume ratio than a reciprocating engine and its shape results in a longer flame travel distance. In addition, the trailing section of the chamber where final combustion occurs is a narrow, wedge-shaped section which tends to quench the flame. At the same time, the charge is swept forward by the moving chamber at extremely high velocity so turbulence is good (6). This allows for a comparatively modest octane number requirement, in comparison to the compression ratio, and a potential for multifuel capability.

Still another area where the Wankel presents a big problem lies in the fact that heat is released more or less continuously in the area near the spark plugs, while one port area of the housing is cooled continuously by the incoming charge (7). The resulting large thermal gradients tend to cause rotor housing warpage. Some problems of this type were experienced during experimentation and will be discussed later.

The future of the Wankel looks very good indeed today since General Motors has purchased the patent rights from Curtiss-Wright and NSU. General Motors plans to make rotary engines optional on Chevrolet Vegas as early as 1974. In addition, Ford Motor Company is looking very seriously at the Wankel at the present time.

The main reasons that the Wankel is growing in popularity are low weight, simplicity, and smooth operation. One misconception that is believed by many people is that the uncontrolled Wankel is lower in emissions than its reciprocating counterpart. The rotary engine is actually worse in HC emissions due to the higher combustion chamber surface to volume ratio. This higher ratio means more cool wall area which tends to quench a flame and produce unburned HC. This higher ratio also reduces peak temperatures and pressures in the combustion chamber resulting in lesser values of NO_x emissions. This fact increases the Wankel's attractiveness since NO_x appears to be the most difficult engine pollutant to abate. Finally, the higher HC emissions in the uncontrolled Wankel do not hurt the engine's popularity since the Wankel lends itself easily to the installment of HC and CO pollution control devices because of its smaller size and simplicity.

Why Natural Gas for a Fuel?

The use of gaseous fuels, principally propane and natural gas, in internal combustion engines is already taking place in many parts of the U.S. primarily to reduce exhaust emissions. Propane is used in trucks and fleet applications, and maintenance costs have been reduced significantly with this fuel since spark plugs last longer and engines can be run for longer distances between oil changes (8). In addition, propane is available at most truck stops throughout the nation.

Natural gas has also been coming into its own recently as an automotive fuel. The fuel can be stored as compressed natural gas (CNG) or liquid natural gas (LNG), which allows for much larger quantities of

natural gas to be stored. Most CNG is 99+ per cent methane (CH_4) and LNG vapor is about 98 per cent methane (9).

In 1970 the Mechanical Engineering Department of Georgia Tech entered a car in the Clean Air Car Race (CACR) which was capable of running on either natural gas or gasoline. The CACR entry was basically a standard 1970 Ford with a 6-cylinder engine and was equipped with a dual-fuel conversion kit, consisting of a venturi mixing valve air intake that enabled natural gas to be used. The special intake bolted to the top of the standard carburetor and switching from one fuel to the other was a very simple matter. CNG was carried in three cylinders in the car trunk at about 2200 psi, giving the car a range of about 100 miles at interstate speeds (10).

The School of Mechanical Engineering also participated in the Urban Vehicle Design Competition in August, 1972, at the General Motors Proving Grounds in Milford, Michigan. The vehicle entered was basically a Mazda R-100 with the stock rotary engine which is being considered in this thesis. The dual fuel conversion was carried out on the Mazda just as it was described previously but only CNG was used in the competition. The exhaust emissions test of the UVDC carried the most scoring weight and the earlier laboratory work done in an effort to minimize emissions for the competition played a large role in this thesis. Most of the pre-competition exhaust emission work was done in the emissions laboratory at Tech with the R-100 engine on a test stand. Some unforeseen problems were experienced with the natural gas powered Mazda and will be discussed later. Even though it did not perform up to its expectation

in the UVDC, the Georgia Tech entry still achieved sixth place in overall performance.

Natural gas has been proven to reduce exhaust emissions, particularly CO and NO_x, in conventional reciprocating engines (9). The special mixing valve or variable venturi used on engines running on natural gas provides a uniform mixture distribution which maintains a constant air-fuel ratio. The gaseous state of the fuel allows for precise fuel metering and a very lean air-fuel mixture to be used and, thus, results in extremely low CO emissions which depend primarily on the air-fuel ratio.

Sizeable reductions in HC have also been found in conventional reciprocating engines operating on natural gas. This fact becomes particularly important if the chemical reactivity of methane is considered. Most of the exhaust hydrocarbons of a natural gas powered engine are methane, and methane has been shown to have very little reactivity in the formation of photochemical smog (9). In addition, the evaporative emissions are almost zero from a dual fuel system car or a car running only on natural gas because the carburetor is dry and no hydrycarbons escape.

Besides the pollutants for which standards have been adopted, other known pollutants are reduced or eliminated by the use of natural gas as a fuel. For instance, lead is totally eliminated since natural gas contains no lead, and soot and carbon deposits are greatly reduced.

A big disadvantage to the use of natural gas as a fuel is power loss. About a 15 per cent loss can be expected when switching from

gasoline to natural gas due to several reasons. Natural gas passes through the intake manifold and into the cylinder as a gas occupying approximately 10 per cent of the available volume, whereas gasoline normally exists as liquid droplets during the intake stroke and takes up very little volume. This reduction in air capacity results in a loss of maximum horsepower of about 10 per cent. Furthermore, the slower flame speed of natural gas lowers peak cylinder pressures and further reduces maximum horsepower (9). NO_x is reduced because of the lower peak cylinder pressures.

Some of this power loss can be recovered by either raising the engine compression ratio or by advancing the spark timing. These procedures could not be followed if the octane rating of natural gas were not so high.

Due to the nature of its gaseous state, natural gas usually offers easy starting (especially in cold weather), reliable idling, and stumble-free acceleration. These improvements in driveability also offset some of the disadvantages of lost peak power.

The concensus of operational experience with natural gas is that 100 cubic feet (LHV=100,000 BTU) are equivalent in average driving to one gallon of gasoline (LHV=114,000 BTU). McJones and Corbeil in (9) state that BTU's/mile with natural gas run from 80 to 100 per cent of the gasoline values, for a 90 per cent average. For extended operation at fairly high speeds the BTU/mile consumption tends to become equal for the two fuels, and 100 cubic feet of natural gas will not equal one gallon of gasoline. During stop and go driving, the lack of an accelerator

pump and a very lean idle mixture with natural gas result in good BTU/mile figures, and 100 cubic feet will better one gallon of gasoline (9).

One major factor that restricts the widespread use of natural gas as a motor fuel is its availability. Natural gas reserves probably could not supply enough gas for automobile demands not to mention the projected problem of conversion to natural gas refueling stations. Areas that do look good for natural gas as a fuel are fleet operations and stationary engines. Many stationary engines in pumping stations and power plants operate today on natural gas.

Finally, much work has been done on the use of natural gas as a fuel in reciprocating internal combustion engines and many of these engines operate today on natural gas. However, the use of natural gas as a fuel in the rotary engine is a relatively new field. It is known that natural gas will allow an uncontrolled Wankel engine to run with cleaner exhaust emissions than will gasoline. But how efficiently do present emission control devices and techniques operate on a natural gas powered Wankel? This question will be answered in part by this study which deals specifically with reducing exhaust emissions from a natural gas powered Wankel which utilizes a catalytic converter and a thermal reactor. Techniques which are employed include variation of engine air-fuel mixture, engine load, engine speed, and exhaust port air injection while measuring the effect of a thermal reactor and catalytic converter on exhaust emissions.

CHAPTER II

DESCRIPTION OF EQUIPMENT

Introduction

The Mazda rotary engine which is the subject of this study was pulled from the car soon after purchase and was mounted on a bench engine-dyno test stand in the emissions laboratory at Georgia Tech. When the work included in this study was begun, the basic engine test setup in the emissions laboratory was essentially the same as it is today. However, some modifications have been made in various areas of the laboratory and will be discussed later. Initial tests were run on the engine, in preparation for the Urban Vehicle Design Competition, and these tests were concerned with exhaust port air injection and thermal reactor efficiency.

When the UVDC was completed in August, the Mazda was brought home and used in a publicity role for its sponsor, Atlanta Gas Light Company, for about a month. Efforts were made in late September to obtain a chassis dynamometer so that engine tests could be carried out with the engine in the car and these efforts finally proved successful. A portable chassis dynamometer was borrowed from Rutherford Equipment, Inc., and the remaining engine emissions tests, which were concerned with the catalytic converter performance, were completed. The portable chassis dynamometer (dyno) was set up in another area of the Mechanical Engineering Research Laboratories and all accessory equipment, including the

emissions instrumentation, was moved from the emissions laboratory to the new site. The test setup and related equipment in the emissions laboratory will be described first followed by the chassis dyno setup.

Bench Engine-Dyno Test Setup

The engine that was tested was a Toyo Kogyo 10A model which came standard in the 1971 Mazda R100 automobile. It was a twin rotor Wankel rated at 110 SAE horsepower at 7000 rpm and it weighed 257 pounds not including the thermal reactor or transmission.

Table 3. Toyo Kogyo 10A Engine Data

Type	Rotary Piston Engine, in Line 2 Rotor, Water Cooled
Displacement	491 cc x 2 rotors (29.96 cu. in. x 2 rotors)
Compression ratio	9.4:1
Compression Pressure	6.0 kg/cm^2 @ 280 rpm (85 lb/in^2 @ 280 rpm)
Maximum Brake Horsepower	100 PS/7000 rpm (JIS) 110 HP/7000 rpm (SAE)
Maximum Torque	13.5 m-kg/3500 rpm (JIS) 100 ft-lbs/4000 rpm (SAE)
Port Timing:	
Intake opens	Primary: 32° A.T.D.C. Secondary: 32° A.T.D.C.
Intake closes	Primary: 40° A.B.D.C. Secondary: 40° A.B.D.C.
Exhaust opens	75° B.B.D.C.
Exhaust closes	35° A.T.D.C.

The engine and dynamometer were mounted to the cement floor of the laboratory on a special engine stand fabricated from steel I-beams. Provisions were made into the stand for small position adjustments.

A Taylor hydraulic dynamometer, Model D-21, with a 6500 rpm maximum operational speed and a dynamometer constant of 6000 was used for loading the engine. Torque and horsepower values were obtained from a Fairbanks-Morse 200-pound capacity beam scale which was fitted to the dynamometer. The dynamometer was coupled to the engine directly with a special driveshaft containing two heavy-duty, automatic-type universal joints. Water was supplied to the dynamometer from the regular building supply through a flexible rubber hose so that torque readings would not be affected. A Watts water pressure regulator was added at the entrance to the dynamometer to keep the load more constant. Exhaust water was directed outside the laboratory through a low restriction 2-1/2 inch pipe.

A specially-built control panel housed operational instrumentation for the Mazda. Stewart-Warner instruments were used exclusively and included a 0-8000 rpm Model 970 tachometer, a 0-100 psi oil pressure gauge, a 140-270°F oil temperature gauge, a 100-250°F water temperature gauge, an ampere gauge, a vacuum gauge of 0-30 inches Hg, and a 10,000 hour hourmeter. A marine throttle was used to control the throttle position with the return spring removed.

The Mazda engine is cooled by water and by oil, which is circulated through the rotors to aid in their cooling. This necessitates a large capacity oil cooler in the car and this was retained in the

laboratory setup. The oil cooler received air from a squirrel cage fan.

The initial cooling water circulation system in the emissions laboratory was a constant source of trouble and eventually caused damage to the engine which will be discussed in more detail later. The engine water was circulated through pipes and into a holding tank of approximately eight gallons in capacity. When the hot water being circulated exceeded a predetermined temperature a thermostat in the outflow pipe opened a solenoid valve and exhausted the water. This external thermostat was usually set for 180°F. The regular internal thermostat, which circulated the water internally until the engine warmed up, was left in place for most of the engine tests. The exhausted cooling water caused the holding tank's level to drop whereupon a make-up float valve added fresh cool water to the system. This system seldom performed satisfactorily due to leaking and sticking float valves, and overheating of the engine occurred occasionally.

The water cooling system was modified in late June and performed much better than the old one. A standard radiator from a RX-2 Mazda was obtained and was mounted outside the laboratory under a shed along with an accompanying fan. Water hoses from the test engine were then run outside to the radiator (a distance of about five feet) and cooling problems were greatly minimized thereafter.

Natural gas was supplied to the engine from the building gas main. The natural gas passed through a dry gas meter, an electric solenoid valve (controlled from the control panel), and a regulator before being directed to the engine. The gas then entered an Impco,

Model 300A mixer which mixed the gaseous fuel with the intake air. This air-fuel mixture could then be controlled by a screw adjustment on the mixer and the mixture remained constant over the wide range of engine operating conditions. The Impco mixer was mounted directly on top of the standard carburetor with the addition of an adapter plate.

All engine exhaust gases were directed out of the laboratory and were passed through a low restriction truck muffler. Fumes were never a problem in the laboratory since the exhaust system was well sealed and a large fan was used for ventilation purposes.

The rotary engine tested comes with an extremely sophisticated emission control system installed on it. First of all the engine has an air injection system, a positive crankcase ventilation system, and an evaporative emission control system. A thermal reactor is standard, also, and will be discussed later. Secondary air is injected by an air pump into the exhaust ports ahead of the reactor. The air pump used is a vane-type pump delivering air at about 1 psi at 900 rpm and 2.5 psi at 2500 rpm. The air is routed to the check valve and the relief valve. The check valve keeps exhaust gas from flowing back into the air pump due to backfire or a drive belt failure. The relief valve releases excessive secondary air when the pressure developed by the air pump exceeds 2.4 psi (11).

This R100 engine has two distributors with two spark plugs in each combustion chamber. The leading spark plugs fire when the rotors are at top dead center of their cycle, and the trailing spark plugs fire at 10° after top dead center (at idle with no vacuum advance). The

firing of the trailing plug is normally controlled by engine speed and coolant temperature but was fired at all times during engine emissions tests.

Other emission control devices on the R100 engine are the anti-afterburn valve, coasting valve, and the balance valve (11). These devices were disconnected during the engine tests because they are ineffective in steady state operation. For this reason, they will not be discussed.

The problem of seal lubrication had to be dealt with when natural gas was used in the engine instead of gasoline. Normally a small amount of oil is delivered by a metering pump to the gasoline in the two float bowls of the carburetor. This oil mixes with the gasoline, burns with the air-fuel charge in the engine, and in the process lubricates the apex and side seals of the rotor (12). This process is not possible with natural gas, of course, because the fuel does not enter the carburetor. Lubrication of the seals is done primarily to reduce friction and thus seal wear due to carbon deposits on the rotor, seals, and housings. To accomplish this purpose approximately 5.5 cubic centimeters of oil is injected normally in ten minutes at 2000 rpm (12). It was decided after due study of the natural gas lubrication problem to see if some oil would still be drawn into the engine through the main carburetor jets if the oil was metered into the dry float bowls. It was proven after operating the engine on natural gas in this manner that oil was actually being drawn into the engine since no appreciable oil build-up was found in the float bowls (10).

Thermal reactor temperatures were recorded during all tests in the emissions laboratory with the use of a chromel-alumel thermocouple that was inserted into the reactor core. Voltage readings were taken from the thermocouple on a Leeds and Northrup potentiometer, Model 8662.

Instrumentation was also available in the laboratory for measuring air-fuel ratios as well. As mentioned earlier a dry gas meter was installed by the gas company for measuring natural gas volumes. Readings were taken in cubic feet of gas at standard conditions of 60°F and 8" W.G. inlet pressure to the meter. Inlet and outlet pressures downstream of the pressure regulator were measured by U-tube water manometers.

Air flows were measured to the engine and to the air pump using Meriam 200 CFM and Meriam 100 CFM capacity laminar flow elements, respectively. The laminar flow elements provide pressure differentials which vary linearly with air flow across a fine honeycomb matrix. A calibration curve is provided and the SCFM air flow can be found once the pressure differential is known.

Chassis Dyno Test Setup

After the UVDC was completed and a portable chassis dynamometer was obtained, further engine emissions tests were run with the engine installed in the car. The portable hydraulic chassis dynamometer was manufactured by Clayton and was capable of handling loads in excess of 200 road horsepower. Horsepower and road speed were measured and displayed on accompanying meters. The dynamometer was loaded by pressing a

convenient rubber lined switch which opened a solenoid valve to let water into the load drum. Another switch opened another solenoid valve which allowed water being used for load purposes to be released from the drum. This system was excellent in that once a load was set, the load remained constant until the engine test was completed.

With the engine being tested in the car, the chassis dyno test setup became a much simpler matter than the bench engine-dyno setup. The stock transmission was utilized and consisted of four forward gears plus reverse. The standard Mazda R100 oil and water radiators were used for cooling purposes with the cooling system having a capacity of 13 U.S. pints without the heater. A 5000 CFM centrifugal fan requiring 2 horsepower from an accompanying electric motor passed air across the radiator for cooling when the car was being tested under loaded conditions.

The tachometer in the Mazda dashboard control panel was used in the tests along with the stock water temperature gauge and ampere gauge. The Stewart-Warner vacuum gauge from the laboratory control panel was used in the car on a specially built mount. A Chrysler stationary engine throttle was obtained and was modified to control the Wankel's throttle position from its underhood mount. The normal accelerator pedal could still be used whenever needed.

Natural gas was supplied to the engine from six cylinders which were mounted in a specially built rear compartment of the car. These gas cylinders were installed for the UVDC, and when full at 2200 psi contain enough gas for about 90-100 miles of moderate to high speed

driving. A pressure gauge was mounted on the Mazda dashboard to show the gas pressure present in the storage tanks. The gas from these storage cylinders was routed through a needle valve to a high pressure regulator which brought the gas down to about 60 psi for transport to the engine compartment. The gaseous fuel next passed into a low pressure regulator which delivered it to the Impco mixer at a pressure of about 4-1/2" water. The gas pressure was measured just in front of the mixer with a magnehelic pressure gauge.

A chromel-alumel thermocouple was used in this chassis dyno setup to measure temperatures at the exit of the catalytic converter. The voltages from the thermocouple were measured with a Hewlett Packard voltmeter which gave a direct readout in millivolts.

Air and fuel flows did not have to be measured in the chassis dyno setup since air-fuel ratios were determined from the exhaust gas composition. This method of calculating air-fuel ratios was not possible in the emissions laboratory since the oxygen analyzer was not available at that time.

A few comparison tests were run with the car on the chassis dyno using gasoline and propane as fuels. A two gallon gasoline can was used for gasoline storage in the car since the regular tank had been removed previously. The fuel was pumped into the carburetor float bowls through a filter by the stock electric fuel pump mounted under the hood.

The propane was stored in an external portable tank. It passed through a pressure regulator, which was heated by engine cooling water to help vaporize the fuel, before reaching the propane mixer. The Algas

propane mixer was similar to the Impco mixer and mounted onto the carburetor in the same manner.

Emissions Sampling and Measurement Apparatus

All of the emissions instrumentation was housed in a sampling train which was used in both the initial and final test setups. The only difference was that an oxygen analyzer was added to the train for testing in the final setup. The following instruments were integrated into the sampling train: a flame ionization detector (FID) for measuring total hydrocarbons, nondispersive infrared (NDIR) instruments for measuring NO_x, CO_2, and CO, and an oxygen analyzer. A diagram of the sampling train is shown in Figure 4. The emissions instruments will now be discussed separately.

Hydrocarbons (FID)

A Beckman No. 400 Hydrocarbon Analyzer automatically and continuously measured the concentration of HC in the engine exhaust gas by utilizing the flame ionization method of detection. The sample passes through a hydrogen flame which is monitored for electrical conductivity. The conductivity of the flame varies according to the sample HC concentration (13).

The fuel used for the hydrogen flame was 40 per cent hydrogen and 60 per cent nitrogen, supplied at 30 psig. Pure air, or zero gas, was also supplied to the flame at 7 psig. Internal pressure regulators for fuel, air, and sample pressure were present on this analyzer. Both fuel and air, as well as all of the span gases, were stored in high pressure cylinders equipped with high pressure regulators.

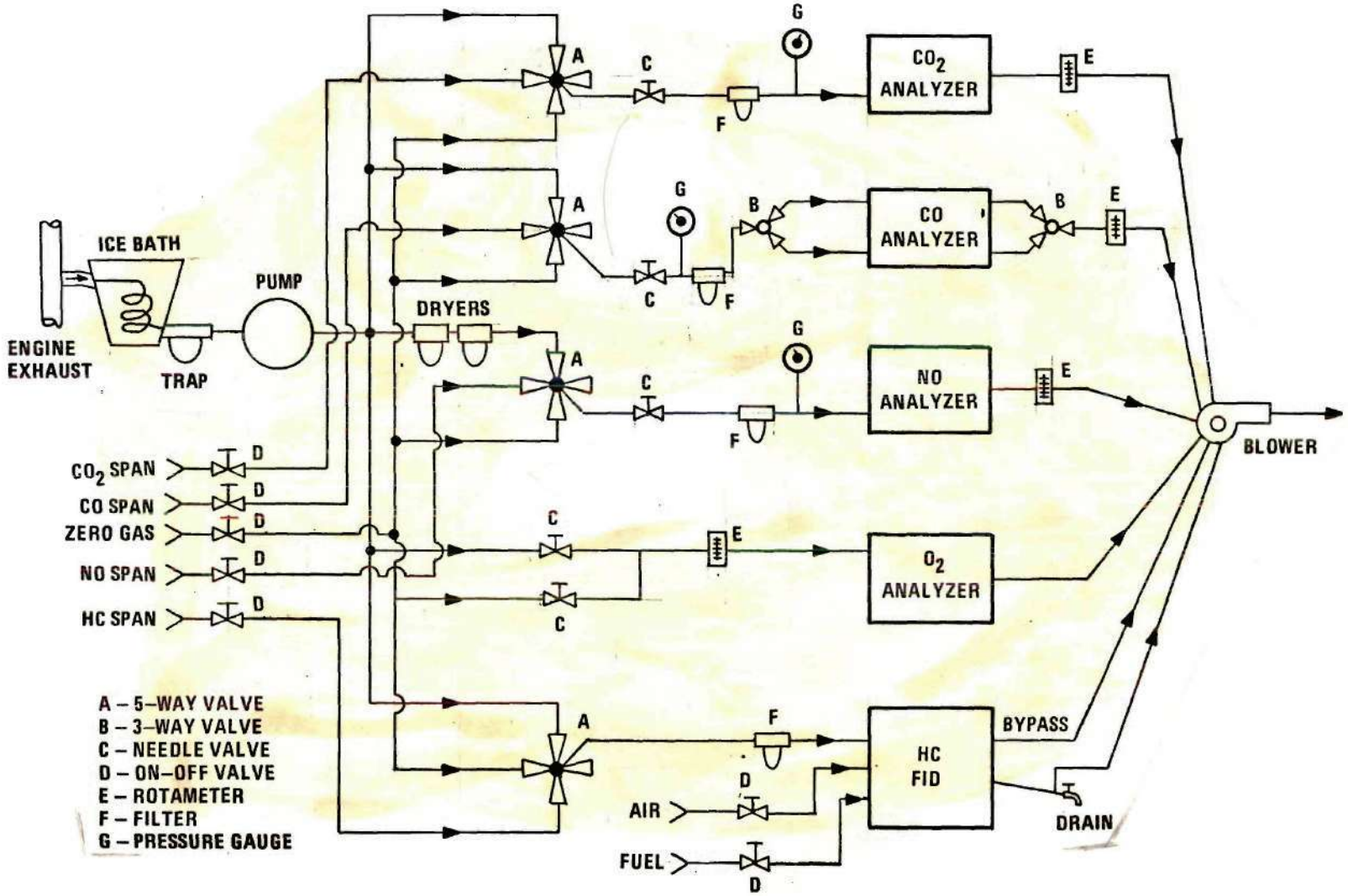

Figure 4. Schematic of Sampling Train

Calibration of the HC analyzer was very simple. Compressed air or zero gas was used to set the zero point and gas with a known concentration of propane (C_3H_8) was used to set the span point. The sample could then be read out directly in ppm by volume since instrument response was linear. However, since most HC emissions measurements are recorded as ppm hexane (C_6H_{14}), the propane concentrations had to be divided by two to give hexane equivalents. The span gas used had a concentration of 970 ppm propane (or 485 ppm hexane). The sensitivity of this FID is accurate to less than 2 per cent hexane on a carbon weight basis.

Nitric Oxide (NDIR)

An Olson-Horiba AIA-2 non-dispersive infrared analyzer was used to monitor nitric oxide (NO). The instrument had two ranges: 0-100 ppm NO and 0-4000 ppm NO. The principle of measurement involves infrared light from identical sources which is passed through two cells, one containing a reference gas and the other containing the sample gas. The infrared beams pass through an optical chopper and into a "gas microphone" detector. A portion of the infrared radiation which passes through the sample cell is absorbed by the specific gas being monitored before it reaches the detector. The important fact is that the percentage of radiation absorbed is proportional to the concentration of the gas of interest in the sample. This percentage is read out on a meter and the meter reading is then applied to a calibration curve which determines the concentration of NO in the sample (14). Compressed air is again used to set the zero point on the meter and the span point is

set using a known concentration of NO in nitrogen. The manufacturer's stated accuracy of this instrument is ±1 per cent, and sensitivity is 0.5 per cent of full scale.

Carbon Monoxide (NDIR)

The instrument used to measure CO was almost identical to the one used for NO. The analyzer section contained two sample cells instead of one, however, and the amplifier section provided three ranges: 0-0.3 per cent CO, 0-3.0 per cent CO, and 0-10 per cent CO. Operation and calibration were virtually the same as in the NO analyzer. One calibration curve was supplied for each range by the manufacturer and a separate span gas (different percentage of CO in nitrogen) was used depending on the sample cell selected. Accuracy and sensitivity specifications are the same as for the NO instrument.

Carbon Dioxide (NDIR)

Carbon dioxide (CO_2) concentrations in the engine exhaust are needed to determine exhaust dilution factors. The instrument used to monitor CO_2 was almost identical to the one used for NO and CO except for two things. The analyzer section contained a different sample/reference cell and the amplifier section provided only one range (0-15 per cent CO_2). This instrument had its own calibration curve also and the span point was set with a known concentration of CO_2 in nitrogen. Accuracy and sensitivity specifications are the same as for the NO instrument.

Oxygen

The oxygen (O_2) concentration in the exhaust needed to be known

in order to calculate air-fuel ratios from an exhaust gas analysis. The instrument used to measure O_2 was a Beckman Model 741 Oxygen Analyzer. The principle of operation is based on the measurement of electrical current developed by an amperometric sensor in contact with the sample (15). The analyzer provides direct readout on a panel meter in percentage O_2 by volume with four ranges available: 0-25 per cent, 0-10 per cent, 0-5 per cent, and 0-1 per cent. No zero point needed to be set for calibration purposes and compressed air was used to set the span point. The accuracy of this O_2 analyzer is ±1 per cent of fullscale or 0.05 per cent oxygen, whichever is greater.

Sample Train System

All five of the sample monitoring instruments were integrated in to one system called a sample train. The instruments were all contained in a formica covered cabinet which also housed all necessary valves, rotameters, and interconnecting tubing. Figure 4 shows the system components in detail. The sample was pulled out of the engine exhaust pipe through a stainless steel probe and through an ice bath by a Model MB-110-10 welded bellows vacuum pump/compressor which was manufactured by the Metal Bellows Corporation. This pump can supply up to 66 SCFH of air if needed.

The ice bath was made up of a coil of 3/8" O.D. stainless steel tubing which was mounted in a large stainless steel bucket. This bucket was filled with ice before any sample was pumped to the instruments. The purpose of the ice bath was to cool the incoming exhaust gas in order to condense most of the water present in the sample. The condensed

water was collected in a trap downstream of the ice bath.

All of the tubing used in the sample train was either stainless steel or teflon, as recommended by the instrument manufacturers. Swagelok fittings were used exclusively in the sample train and provided leak-free connections. All on-off, needle, and five-way valves that were used were stainless steel Whitey valves.

The exhaust sample passed through the pump after going through the ice bath and was then split up among the HC, CO, CO_2, NO, and O_2 analyzers. The HC portion of the sample passed through a Wilkerson 1049 particulate filter before entering the analyzer where most of it was by-passed to the exhaust. The small amount which flowed through the burner was analyzed and then exhausted also. A five-way valve was upstream of the analyzer so that hydrocarbon span gas, zero gas, or sample gas could be routed to each instrument.

The fraction of the sample headed towards the NO analyzer first passed through a drying apparatus consisting of two Wilkerson 4001-2 dryers which contained indicating silica gel. These removed any moisture that may have still been present in the sample after passing through the ice bath. These dryers were needed only for the NO analyzer. After the dryers the sample passed through a needle valve so that the sample flow rate could be adjusted. A Wilkerson 1049 particulate filter was in the line after the needle valve and before the NO analyzer inlet. Inlet sample pressure was measured just ahead of the sample inlet with a Magnehelic 0-10" W.G. manometer. After passage through the analyzer the sample flowed through a Brooks 2-18 SCFH rotameter and on to the

exhaust. Another five-way valve was provided upstream of the needle valve so that NO span gas, zero gas, or sample gas could be routed to the analyzer.

The sample portion routed to the CO_2 meter passed through its own network identical to the NO analyzer network except that the dryers were left out. Also, CO_2 span gas entered the five-way valve instead of NO span gas.

The fraction of sample gas routed to the CO analyzer also passed through a needle valve and a Wilkerson 1049 filter before reaching a three-way valve. This three-way valve allowed for the selecting of one of two sample cells depending on which meter range was desired. Another three-way valve was located immediately downstream of the analyzer to keep sample gas from diffusing back into the sample cell not being used; this would alter meter readings. The sample gas then passed through a Brooks 2-18 SCFH rotameter before being exhausted. Inlet sample pressure was monitored before the first three-way valve with another Magnehelic 0-10" W.G. manometer. Again a five-way valve was present before the needle valve so that CO span gas, zero gas, or sample gas could be routed to the analyzer.

Another fraction of the exhaust sample was directed through a needle valve and a Brooks 2-18 SCFH rotameter before entering the oxygen sensor device of the O_2 analyzer. The sample passed through the O_2 sensor and on to the exhaust. A three-way valve was located before the needle valve which allowed for pure air to be routed to the O_2 sensor for calibration purposes.

On-off valves were provided on the control panel for all zero and span gases. Thus, in order to get span gas to a particular analyzer, the correct span gas on-off valve had to be opened and the correct five-way valve had to be placed in the span position.

Emission Control Devices

Two independent emission control devices were examined and will be discussed in this thesis. They are a thermal reactor which comes standard with the Mazda R100 engine and a catalytic converter which was donated by Engelhard Minerals and Chemicals Corporation.

The thermal reactor is simply a device which further enhances the oxidation of HC and CO to H_2O and CO_2. Oxides of nitrogen are not reduced and may be increased if sufficiently high reactor temperatures result from the combustion of the HC and CO. To achieve a high degree of exhaust system oxidation a high exhaust temperature coupled with sufficient O_2 and residence time to complete the combustion is needed (1). The heat generated by the HC and CO combustion then keeps the reaction going.

Due to its abundance the CO in the exhaust provides most of the combustion generated heat. CO concentrations of several per cent (1 per cent=10,000 ppm) are common in the exhaust gas while HC concentrations are at most a few thousand ppm. As a result thermal reactors are usually developed for rich carburetion. In addition, air needs to be injected into the exhaust ports ahead of the reactor for better oxidation results. This is mandatory for rich air-fuel mixtures and helps with lean air-fuel mixtures up to a point (this will be shown later).

There exists a minimum air injection rate at each air-fuel ratio that provides maximum emission reduction.

An Engelhard PTX Catalytic Converter is the other emission control device that was tested for this thesis. This converter is a platinum catalyst deposited on a ceramic honeycomb structure contained in a stainless steel housing (17). It was placed in the exhaust system immediately downstream of the thermal reactor. The benefit of the catalyst is that it permits the oxidation reactions of HC and CO to occur rapidly at relatively low temperatures. The chemical reaction equation used for the thermal reactor applies to the catalytic converter also except that higher reaction efficiencies (higher specific reaction rates) occur at lower temperatures (16). Catalysts, like thermal reactors, may use air injection to complete combustion or may operate without air utilizing excess oxygen from lean engine mixtures if good heat conservation is achieved.

One problem with catalysts is that they deteriorate with mileage. Lead compounds in exhaust gas resulting from an engine running on leaded fuels (most gasoline presently contains lead) produce deterioration by plugging the converter bed and coating the active catalyst surface. Some catalyst deterioration occurs even using unleaded fuels. A thermal reactor is normally larger than a catalytic converter (as was the case for the devices tested) because it must have a higher residence time to offset lower reaction rates. Both devices have been shown to work well when integrated into an exhaust system of a gasoline powered engine. However, the efficiencies of these devices on engines using natural gas as a fuel have not been previously investigated.

CHAPTER III

PROCEDURE

Introduction

As mentioned previously, testing of the Mazda R100 engine was done in the Emissions Laboratory with the engine on a test stand and in another laboratory with the engine installed in the car. Tests involving the thermal reactor were run in the engine laboratory while catalytic converter testing was carried out primarily in the chassis dyno laboratory. These two testing programs will now be described separately.

Bench Engine-Dyno Test Program

Parameters Varied

Initial testing of the R100 engine involved monitoring the exhaust emissions from the thermal reactor while varying the amount of air injection into the exhaust ports. This was done to find the optimum air injection for minimum CO and HC emissions. The injected air was expressed as a ratio of the air injected into the exhaust ports over the air taken into the engine through the intake manifold.

The amount of injected air was varied at each load and speed with a globe valve placed in the line after the air pump outlet, as shown in Figure 5. The tests were run at engine speeds of 1500, 2500, and 4000 rpm, and at engine loads of 15, 10, and 5 inches Hg intake

manifold vacuum. In addition, data points were taken for air injection ratios of 0, 0.1, 0.2, 0.3, and 0.4 at each of the nine operating conditions.

Air flows were measured with Meriam laminar flow meters as described in Chapter II. Measurement of air injected into the exhaust was not as straightforward as it might seem since different amounts of air were bypassed at different speeds and loads through the relief valve. The system used to obtain this injected air measurement is shown in Figure 5.

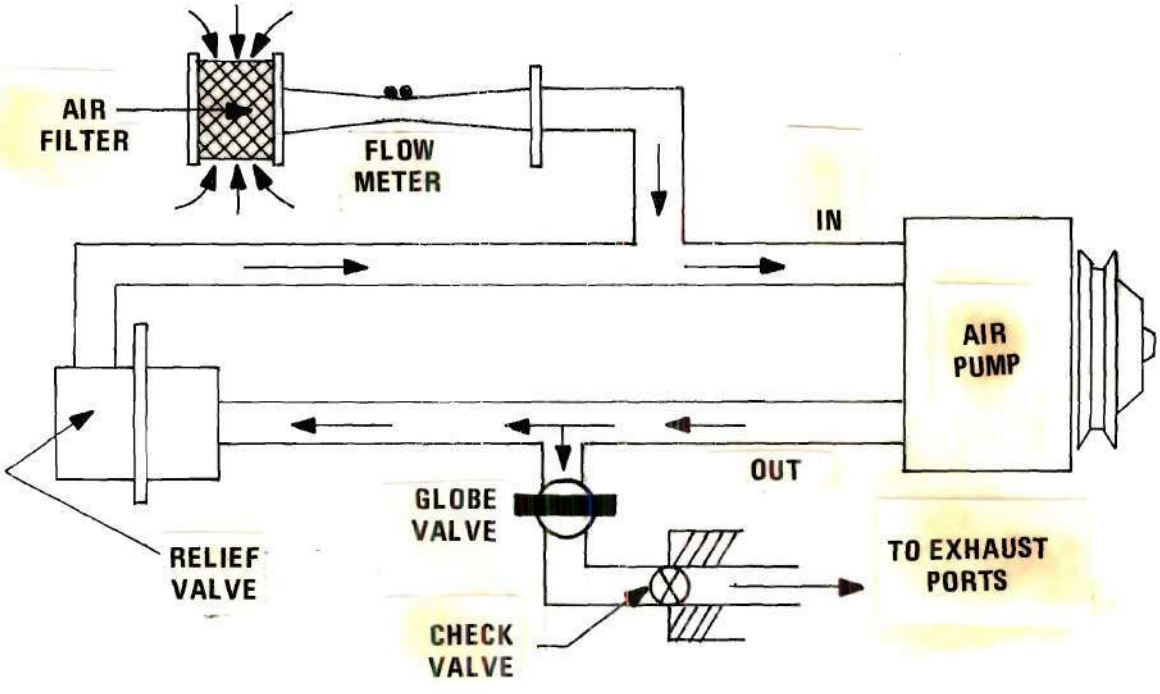

Figure 5. Diagram of Air Injection System

It is shown in the figure that the bypassed air is recycled back to the pump intake so that only the injected air is passing through the flow meter.

The thermal reactor was found after initial tests to be doing poorly at reducing HC emissions. It was then thought that since sufficient reactor temperatures were being recorded and that base HC emissions from the engine were relatively low, perhaps the gases entering the reactor needed to be much richer to enhance the starting of a reaction. For this reason raw methane was injected through a Brooks 2-18 SCFH rotameter into a secondary carburetor where it was mixed with air and injected into the exhaust ports just ahead of the thermal reactor. Exhaust emissions were monitored while injecting 1, 2, 6, and 10 SCFH of methane into the standard flow (with globe valve fully open) of injection air for two test conditions: 1500 rpm with no load, and 2500 rpm with a 10" Hg manifold vacuum. Reactor temperatures were also measured to make sure they were above 1300°F at all times.

Test Procedure

The testing procedures followed in the laboratory will now be described. The first thing that had to be done was warming the instruments up and giving them an initial calibration. The Olson-Horiba instruments were left on all the time once the actual testing began since they required about two hours to stabilize if they were shut off. The Beckman HC analyzer, which required about one hour to stabilize, was shut off overnight to conserve the fuel and air supply.

When all the instruments were warmed up, they were calibrated one at a time beginning with the Olson-Horiba analyzers. One important item when calibrating was to make sure that the inlet pressure and, thus, flow rates of the calibration gases going through the instrument were

the same as the settings made when the exhaust sample gas flowed through the instrument.

The steps followed in calibrating the CO, CO_2, and NO analyzers were essentially the same. First, the respective five-way valve would be turned to "zero," and the static pressure of the flow of zero gas through the instrument of concern was set at 4" W.G. by adjustment of the respective needle valve. The zero point was then set on the instrument with the range selector switch in each position. The five-way valve was then switched to "span," the span gas inlet static pressure was also set at 4" W.G. by the adjustment of the needle valve, and the span point was set on the instrument at each range selection. The five-way valve was then switched to sample and a measurement of the exhaust sample was made with its static inlet pressure also set at 4" W.G.

The Beckman HC analyzer was usually calibrated last. With the fuel pressure set at 30 psig and the air pressure set at 15 psig (on gauges inside the instrument), zero gas was directed from another five-way valve to the analyzer. The sample pressure (also read on an internal gauge) was set at 4 psig with the sample pressure regulator. With these conditions set the zero point was set on the instrument. Next the same pressure conditions were set with the span gas flowing and the desired span point was set. The five-way valve was then set to "sample," and the same pressure conditions were set for the exhaust gas analysis.

While the analyzers were being calibrated, the engine was warmed

up. When the calibrations were completed, the desired engine conditions were set with various readings being taken at each condition. Each engine operating condition was set by varying the dynamometer load and the throttle setting to obtain the desired rpm and manifold vacuum. In addition, the amount of air injection was varied by adjusting the air injection globe valve (shown in Figure 5) at each of the engine operating conditions.

At each of the nine engine operating conditions (3 rpm's × 5 air injection flows = 45 data points), all measurements were recorded. Flow rates of air into the engine and into the exhaust air injection ports were read on the manometer. Room temperature and barometric pressure were recorded. Readings from the four analyzers were recorded next, followed by a torque reading taken from the dynamometer scale. A potentiometer reading was recorded also to obtain reactor temperature from the implanted thermocouple. In addition, a stop watch was used to determine the amount of time taken for five cubic feet of natural gas to flow through the dry gas meter. This gas flow measurement was taken because air-fuel ratios could be calculated from known air and gas flows into the engine.

The laboratory testing that has just been described was run in preparation for the upcoming UVDC as well as for this thesis. Soon after these test runs were completed, a rather serious engine problem occurred. Water was discovered to be coming out of the exhaust ports in substantial quantities. The engine was disassembled completely in order to locate the problem and it was found that both rotor housings

were warped and that there was a crack near the spark plug position in the front housing. Since the engine depends on "O" rings around the housings for water sealing, and since the housings were warped,cooling water was passing into the rotor chambers in these warped areas. The warpage was caused by an overheating of the engine which was credited primarily to a malfunctioning thermostat. The thermostat normally opens when the cooling water heats to about 165°F, cutting off interior engine circulation and allowing external or radiator cooling water to be circulated around the engine. If the thermostat failed to open, the same water would continuously be circulated in the engine by the water pump and would definitely lead to serious overheating problems, especially in high load operation. The tests in the laboratory were often performed with high load engine operation.

Two new rotor housings, a front housing, and a gasket and seal kit were purchased from Mazda Motors of Florida in Jacksonville. The engine was reassembled with these new parts and the water leakage problem was thus solved. No further serious engine problems were encountered.

Chassis Dyno Test Program

Before the engine emissions tests were run using the chassis dyno setup, the Engelhard Catalytic Converter was installed in the Mazda exhaust system just downstream of the thermal reactor. The oxygen analyzer was obtained, in addition, and was integrated into the existing sample train. These were the major differences between the initial and final tests along with the fact that the engine was installed

in the car which was run on the chassis dynamometer.

Parameters Varied

Final testing of the natural gas powered R100 engine involved monitoring emissions from sample points between the thermal reactor and catalytic converter and after the catalytic converter, while varying engine speed, engine load, air-fuel mixture, and air injection. Tests were run at engine loads of 15, 10, and 5" Hg manifold vacuum, at engine speeds of 1500, 2500, and 4000 rpm, and at three air-fuel mixture ratios ($3 \times 3 \times 3 = 27$), thus giving 27 engine operating conditions. At each operating condition, samples from the two exhaust system locations described above were monitored for emissions with full air injection and without air injection. Therefore, there were ($4 \times 27 = 108$) 108 data points where the emissions readings were recorded.

Tests identical to the ones described above were also run on the engine on a limited basis using gasoline and propane as fuels. With these two fuels, tests were run at three engine operating conditions: 1000 rpm and no load, 2500 rpm with 10" Hg manifold vacuum, and 4000 rpm with 5" Hg manifold vacuum. Air-fuel ratios were set once for minimum emissions in all three cases. Measurements were recorded for the four data points for each operating condition, as usual.

In addition, tests were run on the engine to obtain maximum horsepower and torque curves using three different fuels: natural gas, gasoline, and propane. These tests were performed to give good comparisons of Wankel engine power outputs for the three different fuels.

Test Procedure

The testing procedure followed in these final tests described above will now be described. The warming up of the exhaust monitoring instruments and their calibration was done in exactly the same manner as previously described for the laboratory tests. The only difference was that the oxygen analyzer had to be calibrated for these later tests. This was a simple procedure. Compressed air was used to set the span point (no zero point needed to be set) and was supplied to the analyzer sensor at 3 SCFH by the adjustment of a needle valve. The span point was then set for 20.95 per cent O_2. Then the sample gas was supplied to the sensor by the common sample pump at 3 SCFH also (as regulated by another needle valve). Readings were then directly read off the meter in per cent O_2 by volume.

After the instruments were calibrated and the engine was warmed up, the desired engine condition was set. Engine speed was set with the under-the-hood throttle arrangement, which held the throttle steady at the speed desired. The manifold vacuum was set by varying the chassis dynamometer load on the rear tires (the transmission was usually in the highest gear possible for the imposed load to obtain maximum adjustment sensitivity).

Air-fuel ratios were set quite easily by turning a mixture screw located on top of the mixer. The first air-fuel setting during testing was always for the maximum power, the second setting was always for the leanest point of operation before experiencing engine misfire, and the third setting was always for the richest point of operation before

getting misfire. Air-fuel ratios were figured in the later tests from the exhaust gas composition as described by Spindt in (18) and as shown in the Appendix of this thesis.

With a particular engine operating condition established, exhaust composition readings were taken with the sample coming from a point between the thermal reactor and catalytic converter (point 1) with the air injection globe valve closed permitting no flow. Then exhaust composition readings were taken from sampling point 1 with the globe valve opened for full air injection. The globe was then closed and exhaust composition readings were taken with the sample coming from a point after the catalytic converter (point 2). Finally, exhaust composition readings were taken from sampling point 2 with full air injection permitted once again. These readings included exhaust CO, CO_2, NO, HC, and O_2 concentrations along with voltage values from the converter implanted thermocouple. The engine was then ready for a new operating condition.

Emissions measured from a point upstream of the thermal reactor on the natural gas powered R100 engine were obtained at a limited number of operating conditions from a study (10) done on this same engine in March, 1972. These values, however, were treated as guidelines and were not the basis of any conclusions.

The test procedure to obtain the horsepower-torque curves was simple but had to be carried out rapidly to prevent overheating of the engine. A meter which gave road horsepower readings was supplied with the chassis dynamometer and proved quite useful in the testing. First

the dynamometer load was increased until wide open throttle could be achieved without redlining the engine. The engine was then brought down by loading to 1500 rpm at full throttle and a road HP output was recorded. Keeping the engine at full throttle, the engine load was decreased slowly and road horsepower readings were taken at additional engine speeds of 2000, 2500, 3000, 3500, 4000, 4500, 5000, 5500, and 6000 rpm to give 10 good data points. While undergoing these HP tests, the transmission was in second gear. Torque was then calculated from horsepower as shown in the Appendix.

Sources of Error

Errors could have come from many sources in the taking of data for this thesis. These errors probably were the result of equipment malfunctions or irregularities, and some element of human error was involved as well.

Probably the biggest errors resulted from the equipment's inability to hold a set engine test condition. In other words the engine rpm sometimes drifted slightly (±50 rpm) from the desired value during data taking and the engine manifold vacuum would sometimes drift from its initial setting as well. The manifold vacuum and engine rpm stayed more constant in the final tests as compared to the initial laboratory tests because the chassis dynamometer performed much better than the smaller Taylor dynamometer used in the laboratory.

Engine instrumentation could also have contributed to the errors involved. Engine speed was monitored by good quality electrical tachometers which were checked for agreement and were found to differ

by less than 50 rpm. The gauge for measuring manifold vacuum was a good quality Stewart-Warner gauge whose accuracy was better than 1" Hg vacuum.

Variations in laboratory ambient conditions could also have affected exhaust emissions (19) since it was impossible to run all the tests for this study in a single day or on days with the same temperature and barometric pressure. This error was probably small and can be neglected.

The largest number of errors probably resulted from the use of the emissions analyzers. The equipment used was of high quality, but each instrument had at least one minor malfunction which had to be dealt with. The instruments were recalibrated very often to make readings more drift-free. The manufacturer's stated accuracy of the Olson-Horiba instruments was ±1 per cent. The Beckman HC analyzer was accurate to less than 2 per cent and the oxygen analyzer was accurate to ±1 per cent of fullscale, according to the manufacturers.

Errors in emission concentrations could also have resulted from system contamination. Even though filters were replaced often, particulate deposits were still found on some of the system tubing. Contamination would probably result in slightly higher emission readings than were actually present.

Errors could also have been made easily in calculating air-fuel ratios. In the laboratory intake air flows were measured with a laminar flow element which produced a pressure differential. The flow element was of high quality and the manufacturer certified an accuracy of ±0.5

per cent, so significant errors could only have been made in reading the accompanying manometer or in experiencing a leak in the air hose which connected the flow element to the carburetor. Laboratory temperature was measured with a high grade thermometer to the nearest 1°F and pressure was measured with a barometer accurate to .01 inches Hg.

In the laboratory natural gas flow was measured by a dry gas meter supplied by the gas company, and errors may have been experienced here due to human error in reading the meter. The meter was checked with another gas meter, however, after air-fuel ratios calculated from exhaust gas analysis in the final tests were found not to agree under the same engine conditions with air-fuel ratios calculated from actual air and gas flows in the laboratory. It was found at that time that the gas meter in the laboratory was giving flow readings that indicated more gas was passing through than actually was. As a result the air-fuel ratios calculated from exhaust gas analysis in the final tests were used exclusively. Errors in the air-fuel calculations from exhaust gas analysis could have arisen due to inaccuracies in analyzer readings (as discussed earlier) or due to human error in preparing or reading the necessary graphs (as explained in the Appendix).

In conclusion, errors were probably made in many areas of the tests performed for this thesis. It is believed, however, that most of them were relatively small since great effort was put into minimizing errors whenever and wherever possible. Nevertheless, it is a widely accepted fact that emissions tests on automobiles can be quite erratic.

CHAPTER IV

RESULTS

Introduction

After all of the test runs were made, some were taken over again because they showed poor correlation with other points. In addition, many runs were made and then not used directly because there were no significant results to be obtained from their data. However, many meaningful relationships among the parameters were found in the numerous runs that were made. Only data points that lead to the most significant conclusions are included in Chapter IV. All the data points are included in table form in Appendix G.

When an engine is run at air-fuel mixtures other than stoichiometric and/or when air is injected into the exhaust system, Federal test procedures require that corrections for dilution be made so that emissions do not appear to be reduced when in fact concentrations are lower merely due to excess air in the exhaust. Each pollutant (NO, CO, HC) has to be corrected for dilution, and all data presented in this thesis have been corrected. A sample calculation of the correction appears in Appendix C.

Significant Parameter Relationships

Five sets of graphs are displayed in this Chapter to show the results of the tests on the Wankel engine. Each set will be considered

separately and all will be discussed in the following section.

Emissions Variations with Air Injection

The Wankel engine exhaust emissions vary with the amount of air, which is injected into the exhaust ports just upstream of the thermal reactor. This fact is shown in Figures 6 and 7. Data for these curves, as well as those which follow, are included in tables in Appendix G. The emissions in these graphs were obtained from exhaust gas which was sampled from a point just downstream of the thermal reactor (point 1).

Figure 6 shows HC and CO emissions variation as air injection into the thermal reactor is changed. The air-fuel mixture into the engine was kept at approximately 17.5, which was very close to a stoichiometric 17.2. Engine speed was also varied for these tests but engine load was kept constant at 15" Hg manifold vacuum. Timing was set at factory specifications (as described in Chapter II) for every test that was run for this study since it had already been shown (10) that spark timing variations had little effect on exhaust emissions. NO_x emissions are not shown here because they remained constant as air injection varied.

Figure 7 shows HC emissions variation versus the amount of natural gas that was injected along with the standard maximum amount of air (shown in Appendix F) into the exhaust ports just upstream of the thermal reactor. Data for two engine operating conditions is displayed: 1500 rpm with 15" Hg vacuum (low speed and light load), and 2500 rpm with 10" Hg vacuum (moderate speed and load). The air-fuel mixture was kept at 19 for these runs since this was approximately the air-fuel

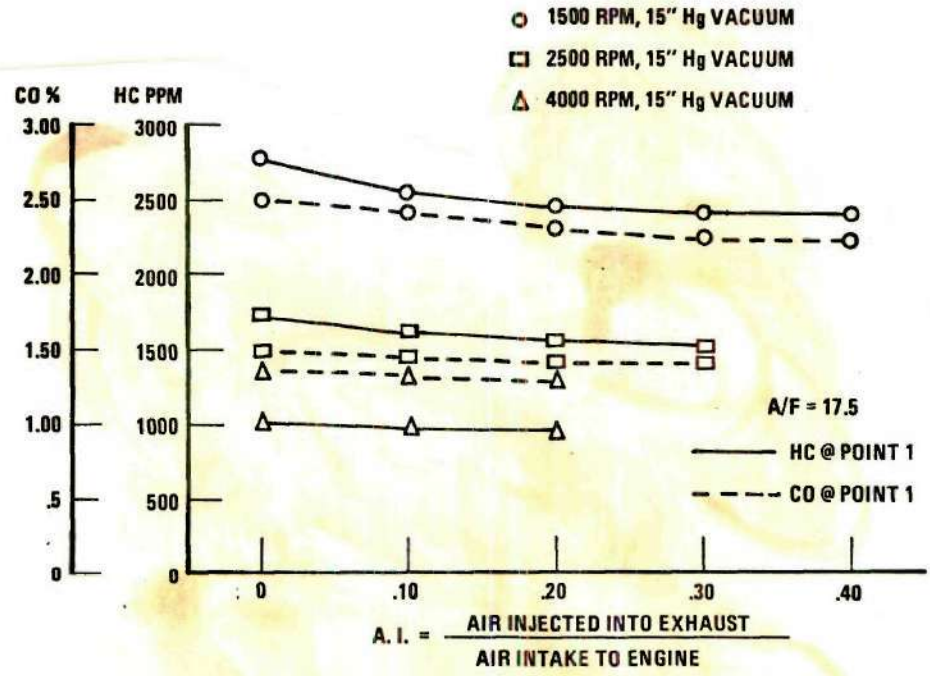

Figure 6. Emissions vs. Air Injection

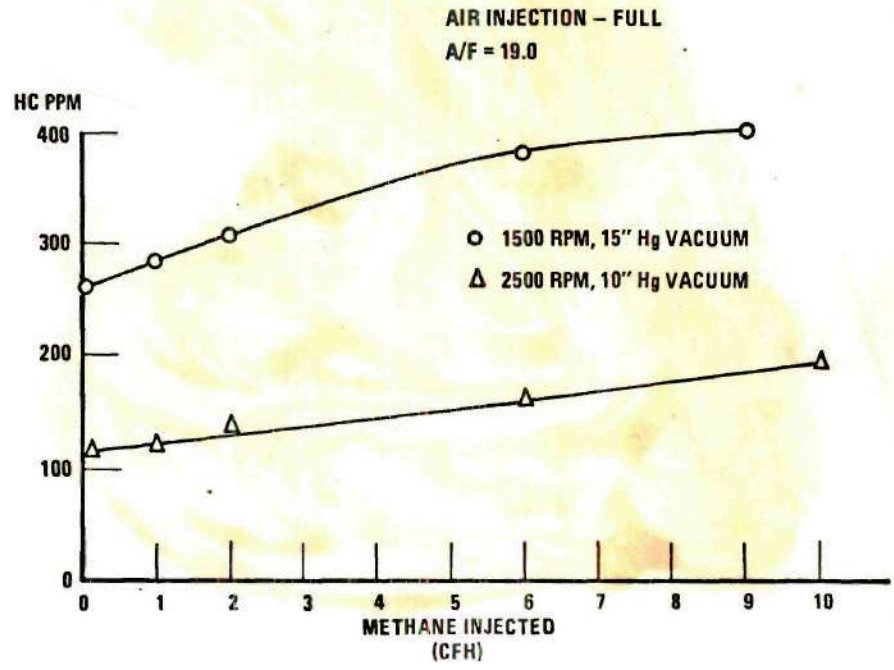

Figure 7. HC vs. Methane Injection

setting which gave minimum HC emissions with no air or gas injection.

HC Emissions Variation with Air-Fuel Mixture

HC emissions are dependent to a large extent on engine air-fuel mixtures. Initially, nine graphs were drawn to display this information, and these graphs represented data for conditions of constant load and constant engine speed, while varying the engine air-fuel mixture. After the graphs were drawn, it was decided that the general trends could be seen with fewer graphs. Therefore, only two graphs, Figures 8 and 9, are included.

Figure 8 shows the variation of HC emissions with air-fuel mixture for three engine speeds (1500, 2500, and 4000 rpm) and a constant load of 10" Hg vacuum. In addition, at each engine operating condition, HC emissions were monitored just downstream of the thermal reactor (point 1) with and without air injection, and just downstream of the catalytic converter (point 2) with and without air injection. Figure 9 shows HC emissions variation with air-fuel mixture for three engine loads (15", 10", and 5" Hg vacuum) and a constant engine speed of 4000 rpm. Here also, at each engine operating condition, HC emissions measurements were made at points 1 and 2, with and without air injection at each point.

CO Emissions Variation with Air-Fuel Mixture

CO emissions are also dependent to a great extent on engine air-fuel mixtures. Nine graphs were initially drawn here also to represent data for conditions of constant load and speed and it was decided that only two graphs were needed to display typical trends.

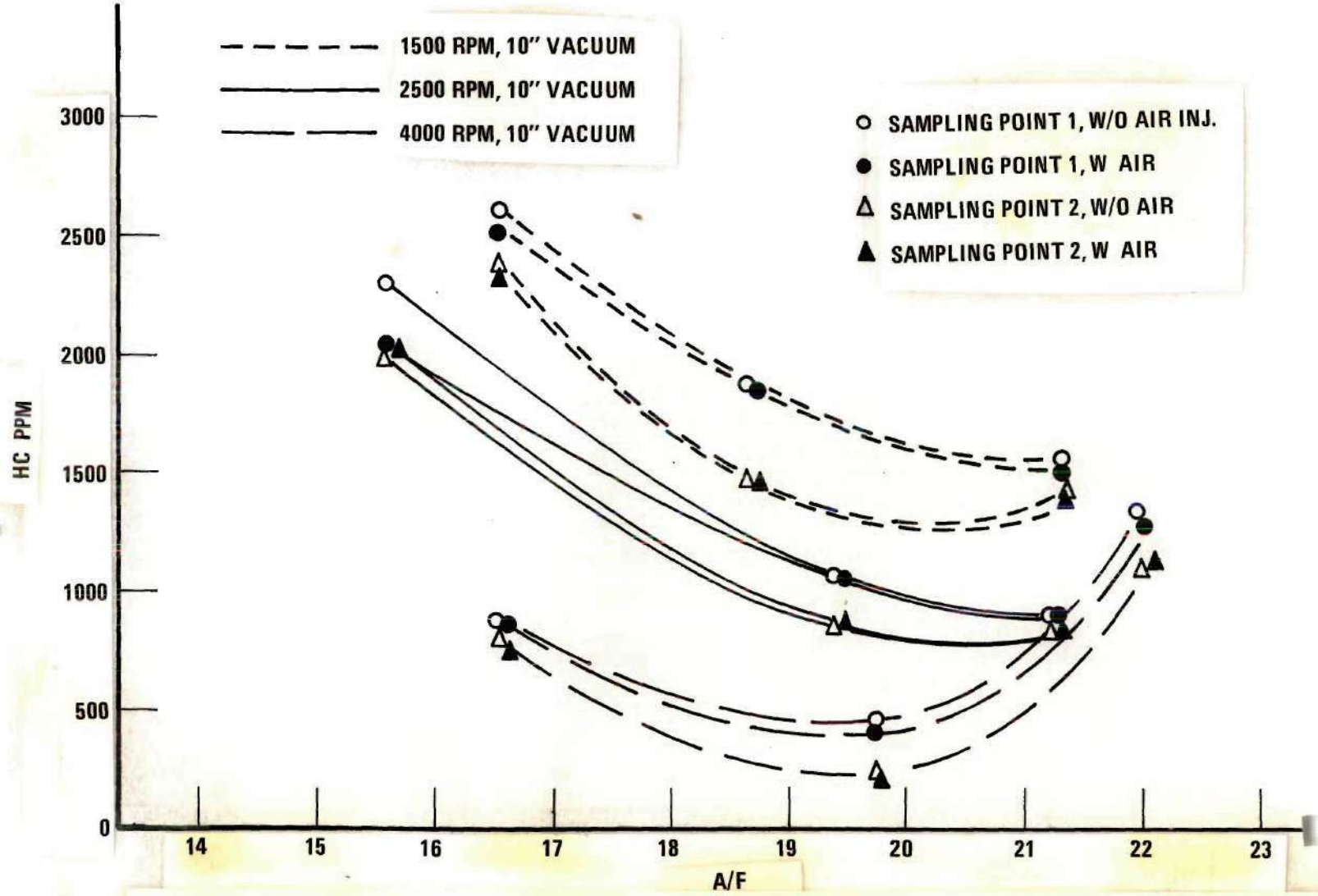

Figure 8. HC Emissions Variation with A/F Mixture and Engine Speed

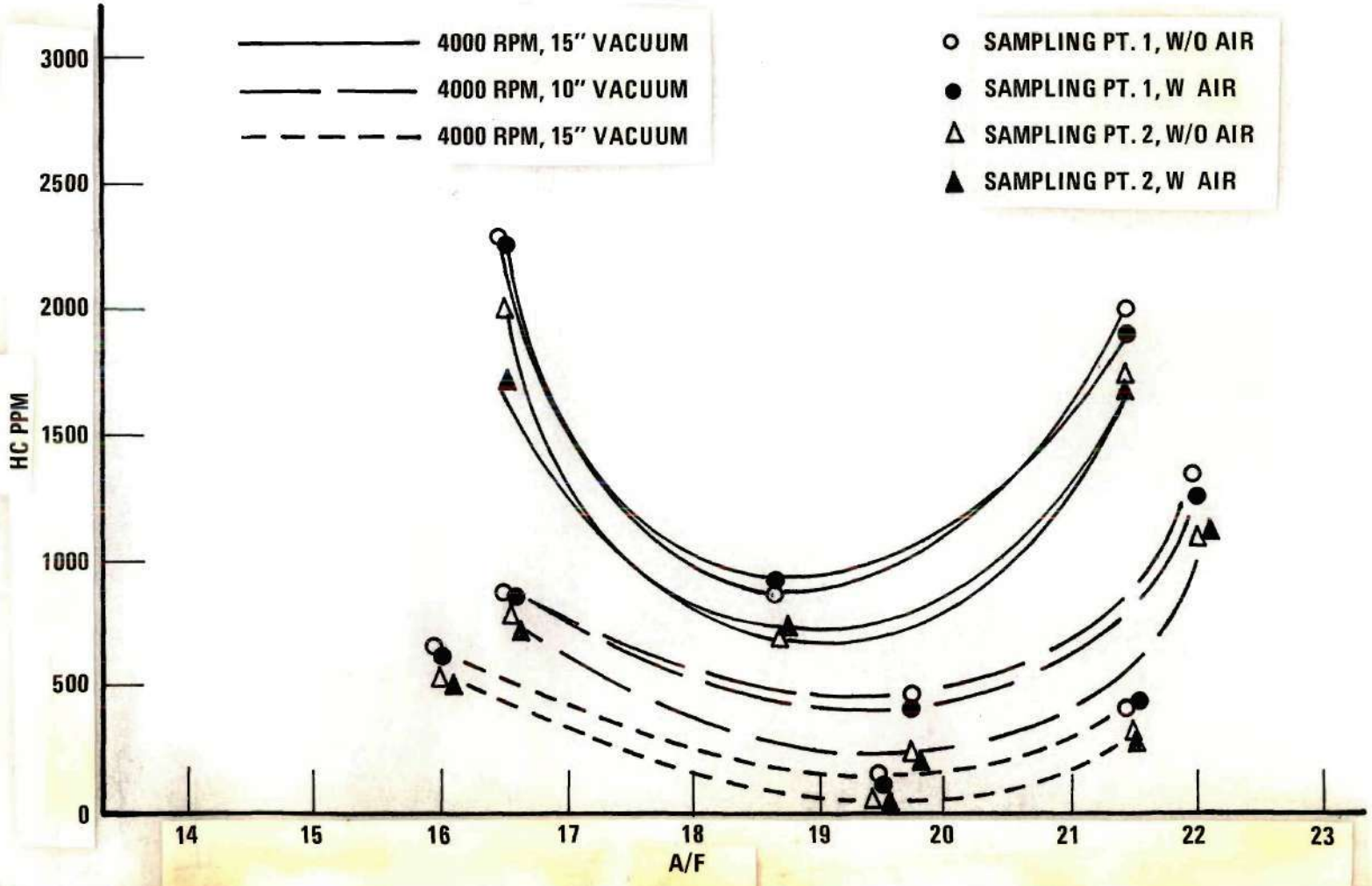

Figure 9. HC Emissions Variation with A/F Mixture and Engine Load

Figure 10 displays the CO emissions variation with engine air-fuel mixture for these engine speeds (1500 rpm, 2500 rpm, and 4000 rpm) and a constant load of 15" Hg vacuum. During these runs, at each engine operating condition CO emissions were sampled at point 1 and point 2, with and without air injection at each point. Figure 11 illustrates the variation of CO emissions with air-fuel mixture for three engine loads (15", 10", and 5" Hg vacuum) and a constant engine speed of 4000 rpm. Exhaust sampling at each operating condition was treated in the same manner previously described.

NO_x Emissions Variation with Air-Fuel Mixture

NO_x, along with HC and CO emissions, are dependent on engine air-fuel mixtures. These original nine graphs have also been reduced to two, Figure 12 and 13, since these two graphs are sufficient to show typical trends.

Figure 12 shows how NO_x emissions are affected by the engine air-fuel mixture at three engine speeds (1500, 2500, and 4000 rpm) and a constant load of 5" Hg vacuum. Figure 13 illustrates NO_x emissions variation with air-fuel mixture for three engine loads (15", 10", and 5" Hg vacuum) and a constant engine speed of 2500 rpm. In both Figures 12 and 13, exhaust NO_x sampling was done at points 1 and 2, with and without air injection at each point.

Emissions Variation with Different Fuels

The exhaust pollutant levels (HC, CO, and NO_x) at the same engine operating condition all varied as the engine fuel was changed. This data is given in Table 4. The engine was run on three different

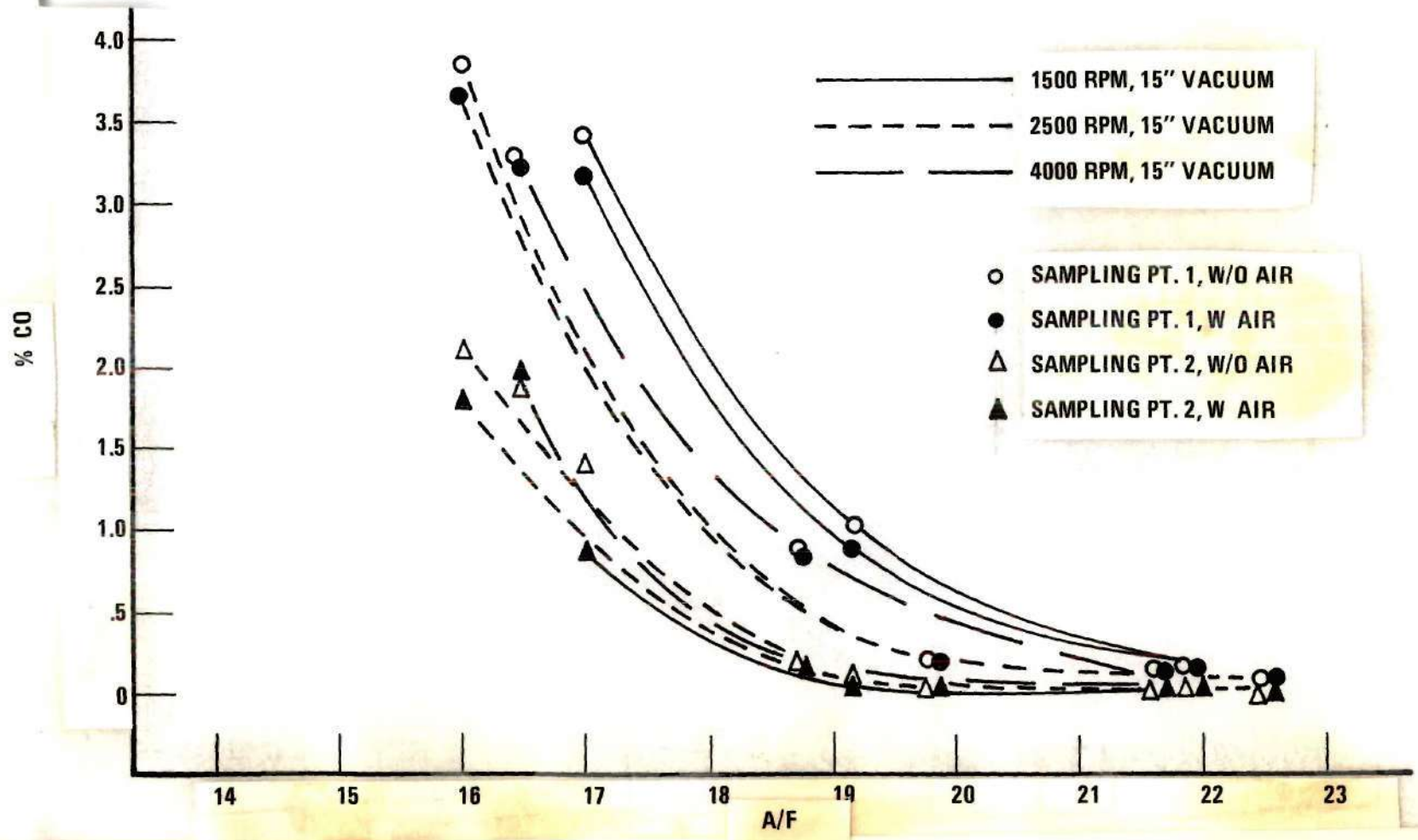

Figure 10. CO Emissions Variation with A/F Mixture and Engine Speed

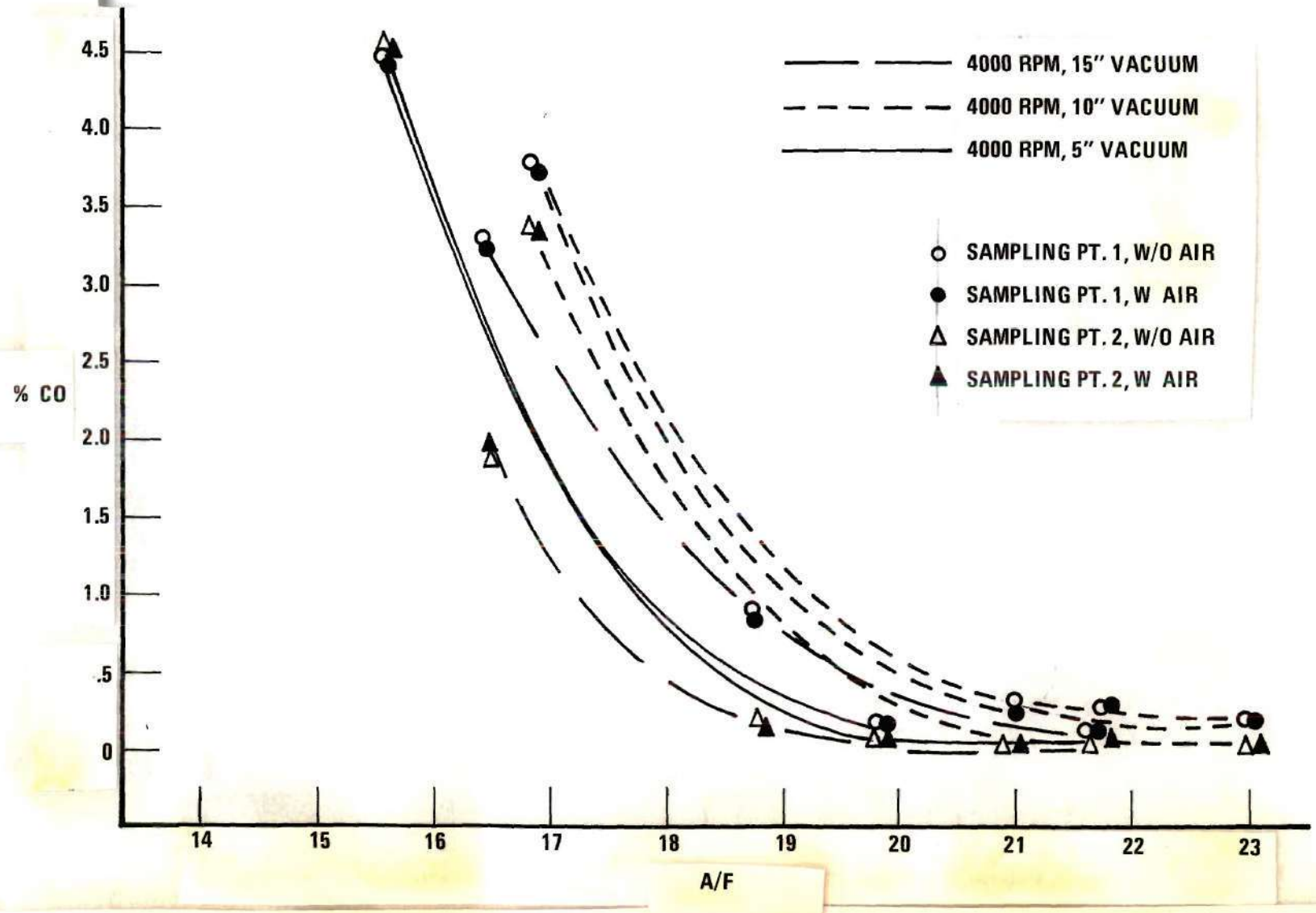

Figure 11. CO Emissions Variation with A/F Mixture and Engine Load

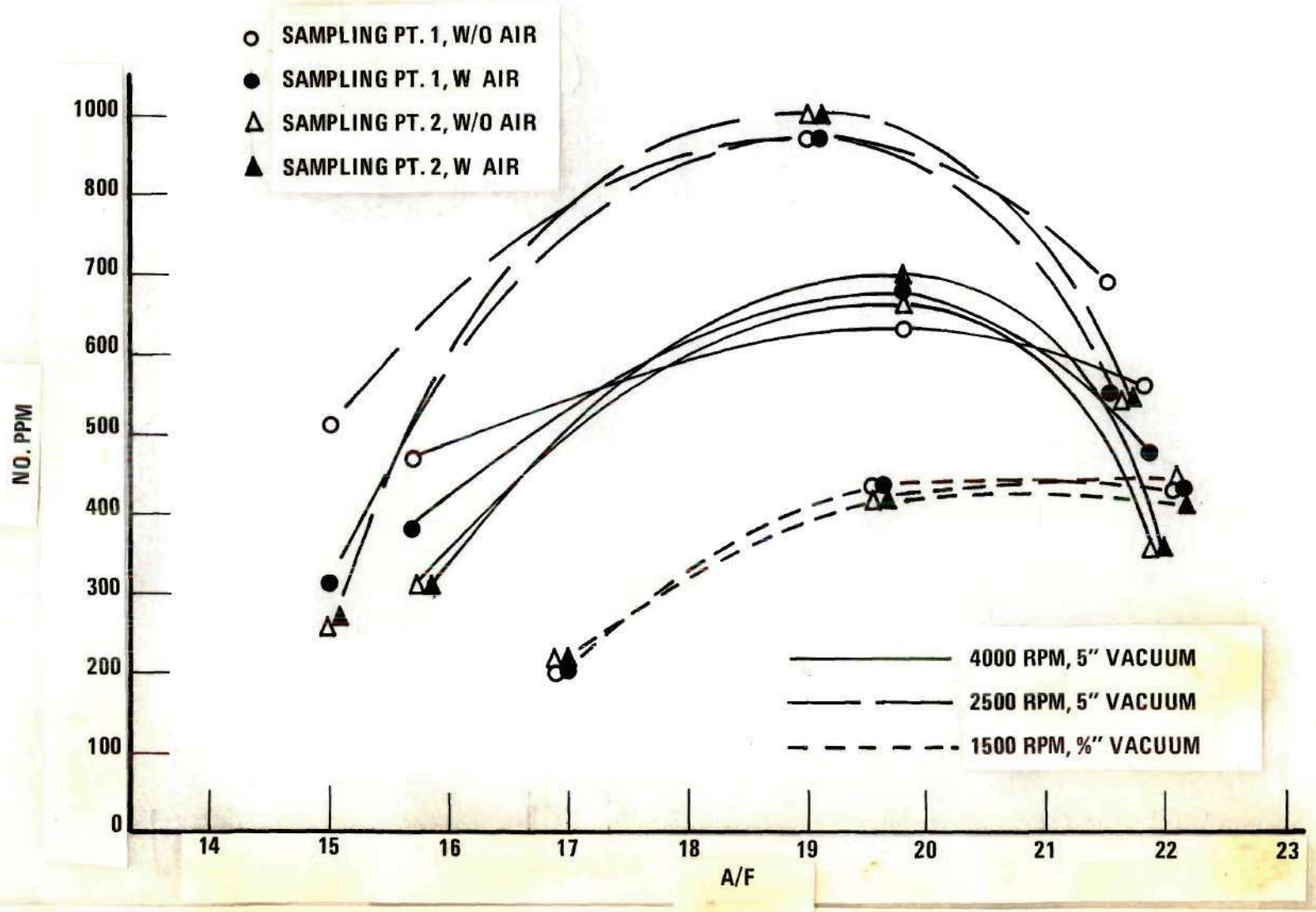

Figure 12. NO Emissions Variation with A/F Mixture and Engine Speed

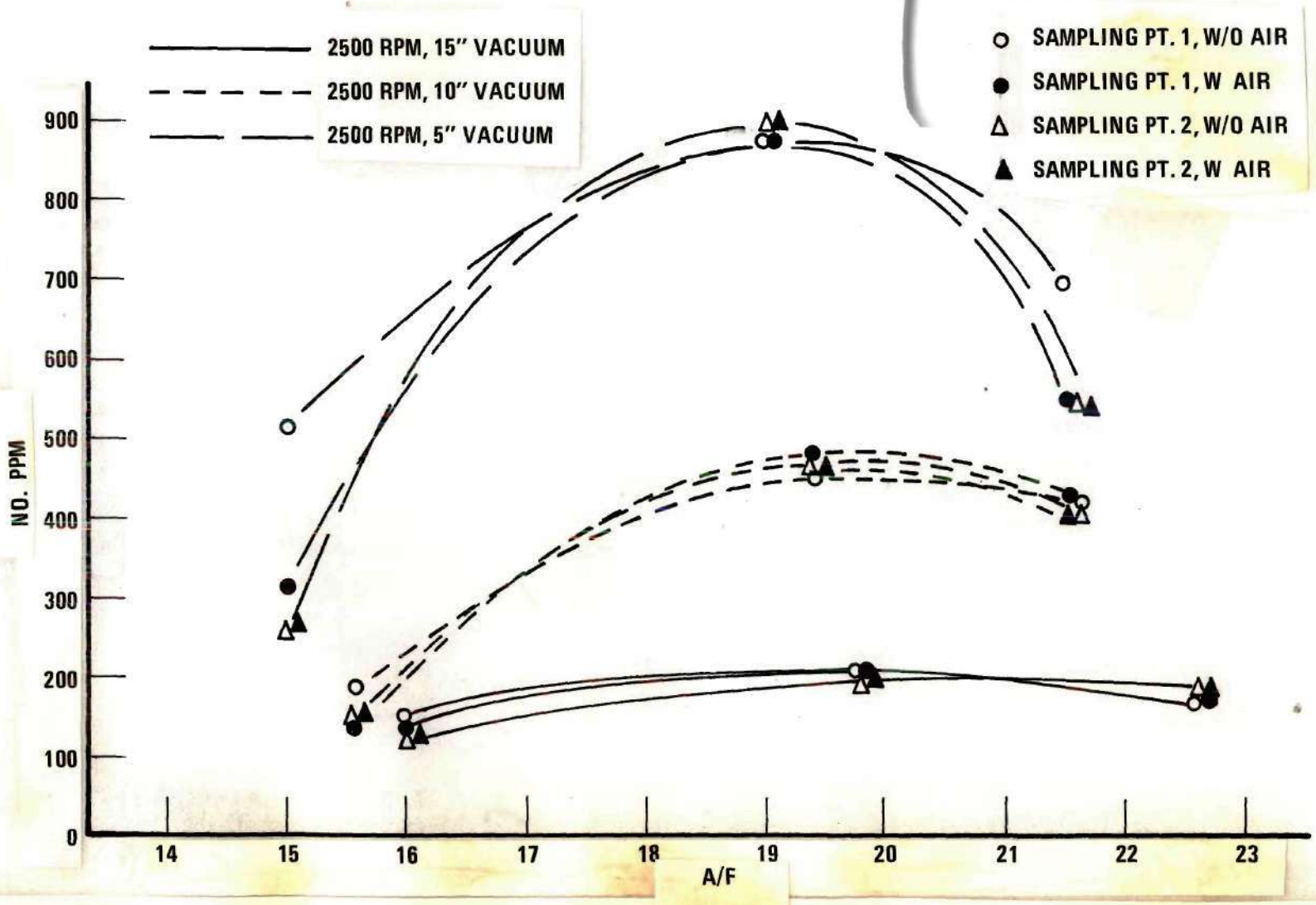

Figure 13. NO Emissions Variation with A/F Mixture and Engine Load

Table 4. A Comparison of Engine Exhaust Emissions with Three Different Fuels with Air Injection

1500 rpm--15" Hg Vacuum		HC	% HC Removed in Converter	CO	NO
Gasoline**	(1)*	4500+ppm	≈45	10+%	220ppm
	(2)*	2900		9 %	220
Propane	(1)	1500		0.53	200
	(2)	970	35	0.05	200
Natural Gas	(1)	1850	16	1.36	100
	(2)	1550		0.07	100
2500 rpm--10" Hg Vacuum					
Gasoline**	(1)	1550		1.80	940ppm
	(2)	630	59	1.10	940
Propane	(1)	515	57	0.31	890
	(2)	220		0.05	890
Natural Gas	(1)	1050	16	1.05	550
	(2)	880		0.22	550
4000 rpm--5" Hg Vacuum					
Gasoline**	(1)	700	55	0.44	1400
	(2)	320		0.20	1400
Propane	(1)	35	49	0.05	1050
	(2)	18		0.02	1050
Natural Gas	(1)	90	33	0.15	700
	(2)	60		0.05	700

*(1) Sampled at point after reactor and before converter.
*(2) Sampled at point after converter.

**Gasoline air-fuel setting not optimized for minimum emissions.

fuels and at three operating conditions (1500 rpm with 15" Hg vacuum, 2500 rpm with 10" Hg vacuum, and 4000 rpm with 5" Hg vacuum), and exhaust emissions were monitored at points 1 and 2 for each operating condition. Full air injection was employed at all times and air-fuel mixtures were set with each gaseous fuel at each operating condition for the minimum HC emission setting. Table 4 gives a good general comparison of not only the exhaust pollutant levels obtained when using different Wankel engine fuels, but also of the catalytic converter efficiencies with the different fuels being used.

Horsepower and Torque Curves with Different Fuels

The last set of graphs is included to illustrate the effects on engine power of switching fuel from natural gas to propane or gasoline. These tests were run with the engine air-fuel ratio set for minimum exhaust HC emissions for each gaseous fuel being used. Timing was set at factory specifications, of course.

Figure 14 shows relative road horsepower levels for the Mazda R100 as they vary with engine rpm for the three fuels that were used. Figure 15 displays the corresponding torque levels developed by the engine on the rear wheels as they vary with engine rpm for each of the three fuels.

Discussion of Results

Emissions Variations with Air and Fuel Injection

It can be seen in Figure 6 that at every engine operating condition displayed, increasing air injection into the exhaust ports continuously decreases both HC and CO emissions. The air injection was

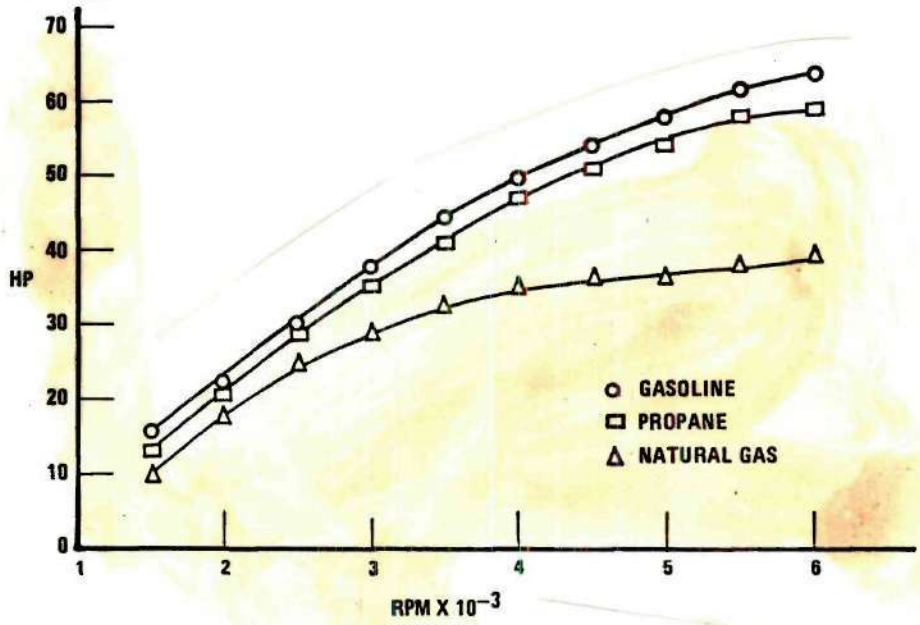

Figure 14. Maximum HP vs RPM

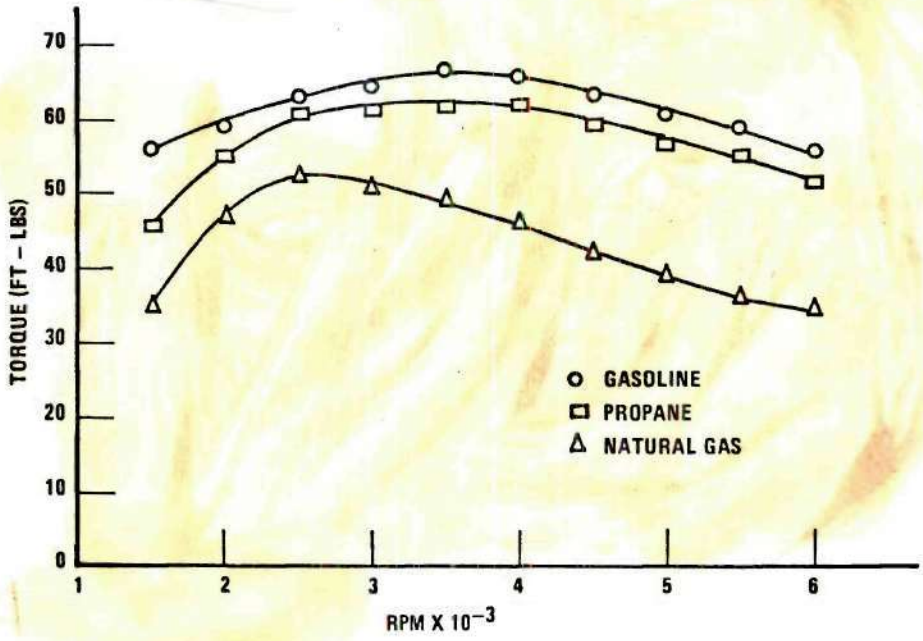

Figure 15. Maximum Torque vs RPM

increased in each case to the limit that could be injected using the standard Mazda R100 air injection pump. It can also be seen here that the HC and CO emissions are decreased very little, but the important fact is that air injection never once, under any of the operating conditions tested, increased HC or CO emissions. This trend shows that the added air always helps by supplying excess oxygen for oxidation processes more than it hurts by reducing temperatures in the exhaust system. NO_x emissions are not shown in Figure 6 since they remain essentially the same under various air injection rates.

Figure 7 illustrates the fact that injecting fuel into the exhaust ports just ahead of the thermal reactor never reduced HC emissions, no matter what engine test condition was run. It was thought that maybe more fuel was needed in the reactor after sufficient temperatures were reached in order to start and sustain a chemical reaction that would oxidize many unburned HC. However, this was not the case and fuel injection only increased HC exhaust emissions. The small amount of reactivity of unburned methane in the reactor will be explained later.

HC Emissions Variation with Air-Fuel Mixture

Figure 8 is typical of HC emissions variation with air-fuel mixture and engine speed. It is shown that HC decreases in every case as the A/F (air-fuel) mixture gets leaner up to a value of approximately 20. After this A/F value is reached, further leaning out of the mixture results in increased HC due primarily to lean misfire and incomplete flame propagation. The HC emissions are higher at richer A/F mixtures

because the oxygen needed for good combustion characteristics is in lesser quantity than in leaner mixtures. Since after-reactions are not very significant when natural gas is used for fuel (this will be discussed more fully later), richer A/F mixtures, which sometimes promote after-reactions if air injection is employed, do not help at all in decreasing HC emissions.

In addition, as engine speed increases under a constant load, HC emissions decrease because turbulent mixing is increased in the combustion chambers, less time is allowed for unburned fuel mixture to leak past the apex seals into the exhaust chamber, and less time is allowed for quenching of the burning charge by the cooler engine walls.

Figure 8 also displays the fact that air injection into the exhaust ports has very little effect on HC emissions from the natural gas powered Wankel. Furthermore, the catalytic converter reduces HC by a small amount here, but the percentage of HC reduction is much less than was previously expected. The reduction is much less than was previously expected. The catalytic converter was more efficient at lower engine speeds due to the fact that residence time in the converter is higher at lower speeds. Air injection has a negligible effect on converter efficiency here, no matter what engine condition is run.

Figure 9 illustrates HC emissions variation with A/F mixture and engine load. It shows, as did Figure 8, that HC emissions are minimized at an A/F ratio between 19 and 20. HC emissions decrease as engine load is increased at a constant engine rpm. This is primarily due to decreased quench thickness in the combustion chamber and increased

combustion temperatures. It is also illustrated in Figure 9 that air injection has very little effect on HC emissions, no matter which sampling point is examined or which engine operating condition is studied. Converter efficiencies are very low here also with the highest efficiency being reached at the condition of lightest load. This fact implies that under lighter loads, increased residence time in the converter overshadows the effect of a small decrease in exhaust gas temperature.

CO Emissions Variation with Air-Fuel Mixture

Figure 10 is typical of all the graphs which show CO emissions variation with A/F mixture. As expected, CO emissions decrease continuously and rapidly with A/F mixture increase (leaner) and reach a very low value near an A/F ratio of 21. It can be seen from Figure 10 that as engine speed varies under constant load, CO emissions follow no pattern. The fact that speed has little effect on CO emissions is expected since oxidation of CO is kinetically limited rather than mixing limited at normal exhaust temperatures. Air injection has a small effect on CO emissions at rich A/F ratios and this fact is expected. A very important result that can be seen from Figure 10 is that converter efficiency in reducing CO emissions, unlike converter efficiency in reducing HC, is very good (averaging about 55% reductions) for all three engine operating conditions tested. The converter efficiency is higher at lower speeds due to longer gas residence time in the converter.

Figure 11 shows again how CO emissions always decrease with an increasing A/F ratio. In addition, as engine load increases with a

constant engine speed, CO exhaust emissions are not affected appreciably on a concentration basis. Air injection has practically no effect on CO emissions once again. Converter efficiency in reducing CO emissions is good (44% reduction) at low load conditions, but efficiencies get worse as load increases due primarily to loss of exhaust gas residence time and less oxygen availability.

NO_x Emissions Variation with Air-Fuel Mixture

Figure 12 is typical of all tests run and shows how NO_x emissions vary with A/F mixture and engine speed. NO_x emissions peak for every operating condition between A/F ratios of 19 and 20. Since NO_x emissions depend upon the maximum combustion temperature and available oxygen concentration in the combustion chamber, this peak is expected. For mixtures leaner than 20, additional increases in NO from increased oxygen concentration are more than offset by lower combustion temperatures and lower formation rates in the lean mixture. With mixtures richer than 19, low peak combustion temperatures and low oxygen concentrations lead to low NO.

NO_x emissions are higher under identical engine loads at speeds around 2500 rpm as shown in Figure 12. These high NO_x emissions at this speed are expected since maximum engine torque is developed around 2500 rpm in the natural gas powered Wankel. This maximum torque results from the maximum cylinder pressures and combustion temperatures. Air injection into the exhaust ports has no effect on NO_x emissions, except for a small increase of NO_x at high load and high engine speed. This injected air probably aided the formation of some NO_x in the thermal

reactor and catalytic converter since exhaust temperatures were highest at this condition. In addition, it appears from Figure 12 that a small amount of NO_x is formed in the catalytic converter when exhaust temperatures are relatively high.

Figure 13 also shows the typical NO_x emissions peak as A/F ratios are increased. It is shown here that NO_x emissions, as expected, are increased as engine load is increased with a constant engine speed. This is the result of higher combustion temperatures which are due, once again, to higher engine torques being developed. Figure 13 also shows evidence that air injection and the catalytic converter each cause a very slight increase in NO_x emissions. The same probably holds true for the thermal reactor since it functions in a similar manner to the catalytic converter.

Emissions Variation with Different Fuels

Exhaust pollutant levels differ a great deal when engine fuel is changed and this fact is represented in Table 4. It is interesting to note in Table 4 that at each operating condition, exhaust HC emissions sampled at both points 1 and 2 are significantly lower when propane is used for fuel. The natural gas HC emissions are always lower than gasoline HC emissions when sampled at point 1, but due to lower converter efficiencies with natural gas, gasoline HC emissions are lower at one engine condition when sampled at point 2 than are methane HC emissions. In every case it can be seen that converter efficiencies are higher for gasoline HC with propane second and methane last. Hence, even though methane HC emissions are lower than gasoline HC emissions (10) and about

equal to propane HC emissions in the exhaust gas which enters the thermal reactor, low reactor and converter efficiencies in reducing natural gas HC emissions nullify most of its initial gains.

CO emissions were higher using gasoline and lower using propane in every case studied. The fact that much lower CO emissions were obtained when using gaseous fuels was due primarily to the leaner A/F mixtures being run. NO_x emissions, as expected, were much lower when burning natural gas than when burning the other two fuels. It is interesting to note, however, that the NO_x emissions when using propane were closer to those obtained when using gasoline (highest NO_x emissions) than when using methane.

Horsepower and Torque Curves Using Different Fuels

Figure 14 shows, as expected, that maximum engine horsepower is developed when gasoline is used for fuel. Horsepower levels obtained when using propane are about 7 per cent less than those obtained when using gasoline. Natural gas horsepower readings leveled off rapidly at about 3500 rpm and resulted in an average horsepower reduction of 29 per cent as compared to gasoline horsepower levels. This is more than was previously expected.

Figure 15 shows maximum torque variation with engine rpm for the three different fuels. Gasoline torque readings are highest, of course, since the gasoline horsepower levels are highest at the respective engine speeds. It should be noted, however, that maximum engine torque developed using natural gas peaks out around 2500 rpm while torque peaks when using propane and gasoline occur around 3500 rpm. This last

fact is due primarily to the slower flame speed experienced in a natural gas mixture.

CHAPTER V

CONCLUSIONS

The results given in Chapter IV lead to several important conclusions about the natural gas powered Mazda R100 rotary engine. First of all, hydrocarbon and carbon monoxide emissions were minimized at all engine operating conditions when the maximum amount of air was injected into the exhaust ports just upstream of the stock thermal reactor. In addition, natural gas injection along with the full air injection into the reactor, after sufficient reactor temperatures had been reached, did not result in a hydrocarbon concentration decrease in the reactor exhaust gases.

Hydrocarbon emissions were minimized from the natural gas powered Wankel when the engine air-fuel mixture was between 19 and 20. Hydrocarbon emissions also decreased as engine speed increased and as engine load increased. Neither the air injection nor the special emission control devices utilized in the exhaust system (thermal reactor and catalytic converter) resulted in a significant decrease in HC emissions.

Carbon monoxide emissions continually decreased as the engine air-fuel mixture was made leaner, and variations in engine speed and/or engine load did not have a substantial effect on CO emissions. Air injection never lowered CO emissions appreciably but the catalytic converter resulted in a significant decrease in CO emissions (averaging around a 70 per cent reduction).

Nitric oxide emissions were maximized when the engine air-fuel mixture was between 19 and 20 and when the engine speed was approximately 2500 rpm, the speed where maximum engine torque was developed. Increasing engine load also increased NO_x emissions.

Final exhaust HC emissions obtained when using natural gas for fuel were generally lower than HC emissions when burning gasoline and always higher than HC emissions when burning propane. It is important to note, however, that air-fuel mixtures were not optimized for minimum emissions when gasoline was used for fuel. Had this been the case, gasoline HC emissions would probably have been much lower. Final CO exhaust emissions resulting from natural gas use were markedly reduced from CO emissions with gasoline and slightly increased from CO emissions with propane. Final NO_x exhaust emissions from natural gas use were substantially lower than those resulting from gasoline or propane use.

The efficiency of the catalytic converter in reducing HC emissions was always higher when burning gasoline or propane as compared to natural gas. In fact, this higher oxidizing efficiency on gasoline could probably result in consistently lower final HC emissions on gasoline.

The use of natural gas for fuel in the Wankel resulted in an average maximum horsepower loss of 29 per cent as compared to gasoline use and 22 per cent as compared to propane use. Natural gas also shifted the speed at which the maximum engine torque was developed from 3500 rpm, as is the case when gasoline and propane are used, to 2500 rpm.

Thus, various conclusions have been reached as to how to minimize exhaust HC and CO emissions from the natural gas powered Wankel. NO_x emissions, even at their peak conditions, have been shown to be relatively low when burning natural gas. The major disadvantage, however, to the use of natural gas for the purpose of reducing exhaust emissions is the fact that present exhaust emission control devices, such as thermal reactors and catalytic converters, are relatively inefficient in cleaning up exhaust HC emissions which result from the use of natural gas.

CHAPTER VI

RECOMMENDATIONS

Testing that was performed on the Mazda R100 engine was much simpler and easier to accomplish when it was carried out with the engine in the car rather than when the engine was on a test stand. This was due primarily to the good quality and ease of operation of the portable chassis dynamometer that was borrowed for the tests. It is for these reasons that the purchase of a chassis dynamometer, either portable or permanent, is recommended if further automobile engine work is to be carried out. The sampling train used in these tests was already overcrowded and became even more so with the addition of an oxygen analyzer. It would greatly improve the appearance of the sampling train, as well as its ease of maintenance, if a larger model was constructed.

In reference to further work on the Wankel, it would be interesting to put the engine through various tests while using propane as fuel and measuring corresponding exhaust emission levels. From this study propane appeared to be an excellent fuel to use to reduce Wankel engine HC exhaust emissions.

Hopefully, more engine exhaust emission tests of some kind will be run in the future to take advantage of the excellent equipment that is available at Tech for the continuous monitoring of HC, CO, CO_2, NO, and O_2 concentrations in a sample gas.

APPENDIX A

SAMPLE CALCULATION OF AIR-FUEL RATIO

(LABORATORY METHOD)

The following raw data have been obtained:

Air flow manometer reading:	0.607 inches
Initial gas meter reading:	0 cubic feet
Final gas meter reading:	10 cubic feet
Time between meter readings:	2:42
Laboratory room temperature:	80°F
Barometric pressure:	29.45 inches Hg

Step 1

The calibration curve for the Meriam laminar flow element is entered for a pressure differential of 0.607 inches and the following is obtained:

Air Flow - 32.4 SCFM

Step 2

The above air flow is in standard cubic feet per minute (SCFM) and must be corrected for temperature and pressure variations. The flow element is calibrated at a temperature of 70°F and barometric pressure of 29.92 inches Hg. Correction tables are included in the instruction manual and the following values are given:

80°F gives a correction factor of 0.9674.
29.45 inches Hg gives a factor of 0.9843.

Therefore:

Actual Flow (CFM) = (32.4 SCFM)(0.9674)(0.9843)

Actual Flow = 30.8 CFM

Step 3

The mass flow rate of air must be obtained next. This is done by multiplying the above by the air density. From the perfect gas law:

$$\rho_{air} = 0.0735 \text{ lbm/ft}^3$$

Therefore:

$$\dot{m}_A = \text{Mass Flow Rate of Air}$$

$$\dot{m}_A = (30.8)(\rho_{air})$$

$$= (30.8)(.0735)$$

$$= 2.26 \text{ lbm/min}$$

Step 4

The volume flow rate of natural gas is determined next.

$$\text{Total Volume} = 10 - 0 = 10 \text{ ft}^3$$

$$\text{Total Time} = 2{:}42 = 2.7 \text{ min.}$$

$$Q_{NG} = \frac{10 \text{ ft}^3}{2.7 \text{ min}} = 3.8 \text{ ft}^3/\text{min.}$$

This flow rate must be corrected for temperature, since the dry gas meter is calibrated at 60°F:

$$Q_{NG_{corrected}} = (3.8 \text{ ft}^3/\text{min})\left[\frac{540°R}{520°R}\right]$$

$$= 3.95 \text{ ft}^3/\text{min}$$

Step 5

The mass flow rate of natural gas must now be obtained. The density of natural gas at room temperature is also found from the perfect gas law:

$$\rho_{NG} = 0.0415 \text{ lbm/ft}^3$$

Therefore:

$$\dot{m}_{NG} = \text{Mass Flow Rate of Natural Gas}$$

$$= (Q_{NG_{corrected}})(\rho_{NO})$$

$$= (3.95 \text{ ft}^3/\text{min})(0.0415 \text{ lbm/ft}^3)$$

$$= 0.164 \text{ lbm/min}$$

Step 6

Finally the air-fuel ratio can be found by dividing the mass flow rate of air by the mass flow rate of fuel:

$$A/F = \dot{m}_A/\dot{m}_{NG}$$

$$= (2.26 \text{ lbm/min})(0.164 \text{ lbm/min})$$

$$A/F = 13.78$$

APPENDIX B

SAMPLE CALCULATION OF AIR-FUEL RATIO FROM EXHAUST GAS ANALYSIS (18)

The following raw data have been obtained:

Concentration of CO_2 in exhaust gas:	10.55	per cent
Concentration of CO in exhaust gas:	1.0	per cent
Concentration of O_2 in exhaust gas:	4.2	per cent
Concentration of HC in exhaust gas:	930	ppm

Step 1

The FID HC reading must be converted to a percentage concentration of HC (per carbon atom). The FID reading measures ppm of HC as n-Hexane, which is C_6H_{14}.

$$\text{Per Cent HC (Per Carbon Atom)} = P_{CH} = 6(930)(10^{-4}) = 0.558$$

Step 2

$$\text{Let } R = \text{per cent CO/per cent } CO_2 = P_{co}/P_{co_2} = 1.0/10.55 = 0.095.$$

Step 3

$$\text{Let } Q = P_{O_2}/P_{CO_2} = 4.2/10.55 = 0.398.$$

Step 4

$$F_b = (P_{CO} + P_{CO_2})/(P_{CO} + P_{CO_2} + P_{CH})$$

$$= (1.0 + 10.55)/(1.0 + 10.55 + 0.558)$$

$$= 0.955.$$

$$F_c = f(\text{fuel}) = .75 \text{ for natural gas.}$$

Step 5

Then:

$$A/F = F_b\left[11.492\ F_c\left(\frac{1+R/2+Q}{1+R}\right) + \frac{120(1-F_c)}{3.5 + R}\right]$$

$$= .955\left[11.492(.75)\left(\frac{1+0.047+0.398}{1+0.095}\right) + \frac{120(0.025)}{3.5 + 0.095}\right]$$

$$= 18.7$$

A program of the previous steps was prepared on the Wang Calculator and was used to obtain numerous data points so that two graphs could be constructed. These graphs made A/F ratio determinations fairly simple and quick when the Wang Calculator was unavailable and they are shown in Figures 16 and 17.

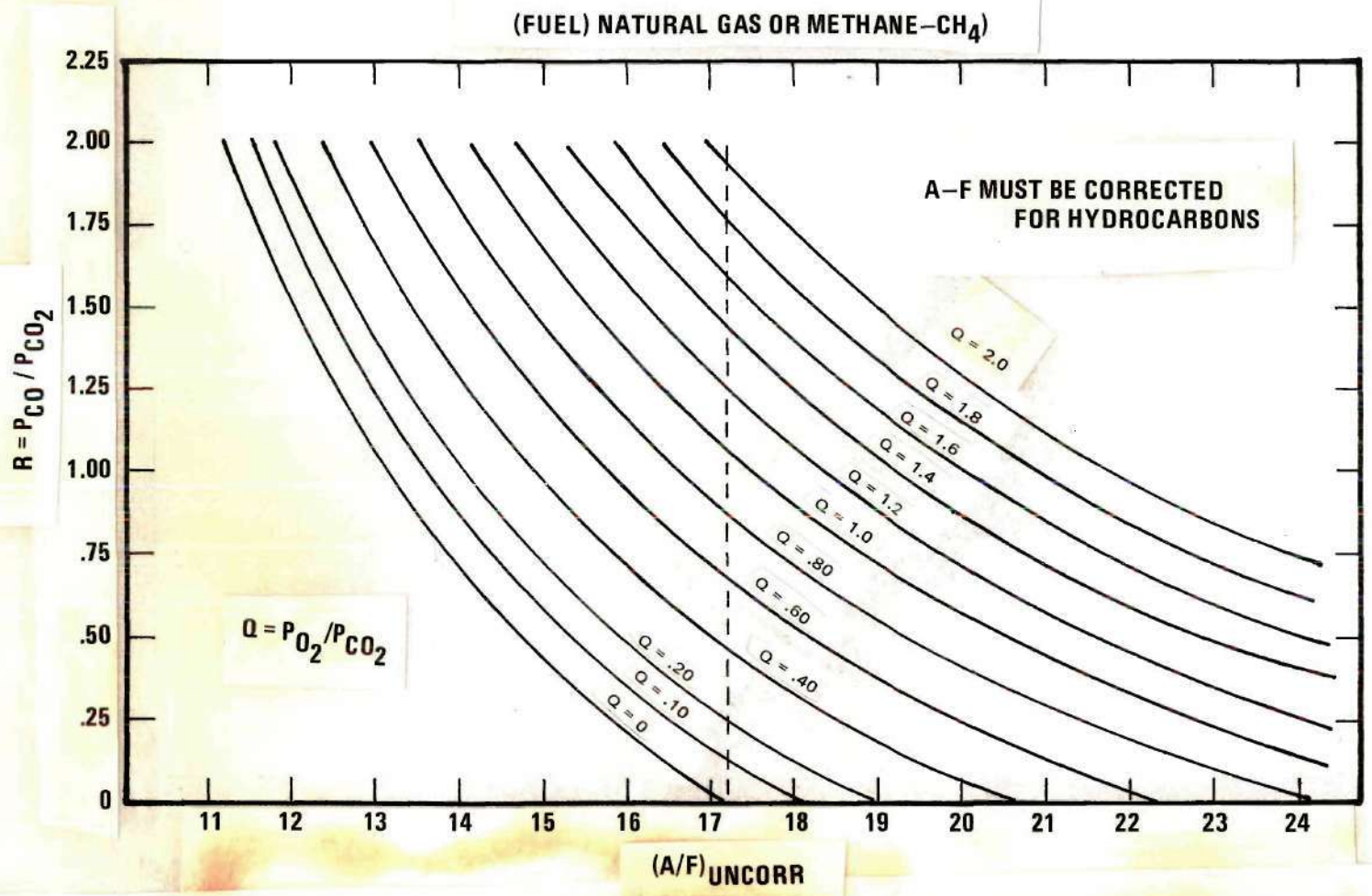

Figure 16. Air-Fuel from Exhaust Gas Plot

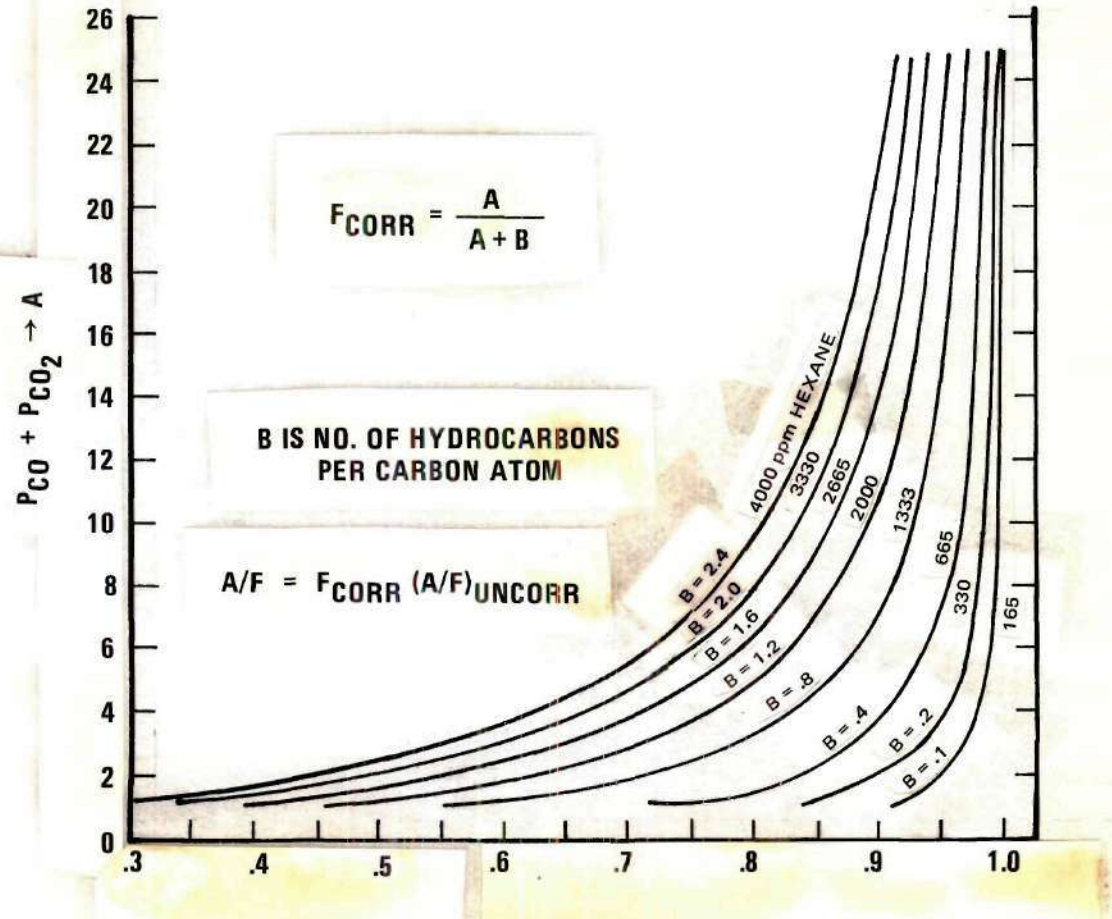

Figure 17. HC Correction Factor

APPENDIX C

CORRECTION FOR EXHAUST DILUTION

Federal regulations specify calculations for corrections of exhaust gas concentration inaccuracies due to dilution. For natural gas, the following formula is used:

$$\text{C.F. (Correction Factor)} = \frac{11.0}{\%CO_2 + (0.5)\%CO + (6)\%HC}$$

An example will be shown using the following data:

$CO_2 = 9.7\%$

$CO = 1.35\%$

$HC = 1830 \text{ ppm} = 0.1830\%$

Therefore:

$$\text{C.F.} = \frac{11.0}{9.7 + (0.5)(1.35) + 6(0.1830)}$$

$$= 0.962$$

The numbers obtained for the pollutant concentrations must now be multiplied by this correction factor to obtain corrected values:

$$NO_{corr} = 0.962(NO) = 0.962(475) = 457 \text{ ppm}$$

$$HC_{corr} = 0.962(HC) = 0.962(1830) = 1760 \text{ ppm}$$

$$CO_{corr} = 0.962(CO)$$

$$= 0.962(1.35) = 1.30\%$$

APPENDIX D

SAMPLE CALCULATION OF TORQUE VALUES

The torque values were calculated directly from the road horsepower values:

Road Horsepower Developed: 55

@ RPM of: 5000

$$HP = \frac{2\pi(Torque)(RPM)}{33,000}$$

Therefore:

$$Torque = \frac{HP(33,000)}{6.28(RPM)} = \frac{55(33,000)}{6.28(5000)}$$

$$= 58 \text{ ft.lbs.}$$

APPENDIX E

ENGINE INTAKE AIR VERSUS RPM AND LOAD

A/F ≈ 17.2 (Stoichiometric)
Fuel = Natural Gas
Air Temperature ≈ 75°F

Air Intake (SCFM)	RPM	Load (Vacuum)
24.3	1500	5
18.5	1500	10
12.8	1500	15
36.7	2000	5
25.6	2000	10
19.1	2000	15
58.5	3000	5
45.1	3000	10
31.0	3000	15
82.0	4000	5
61.5	4000	10
41.7	4000	15
102.0	5000	5
77.5	5000	10
54.0	5000	15

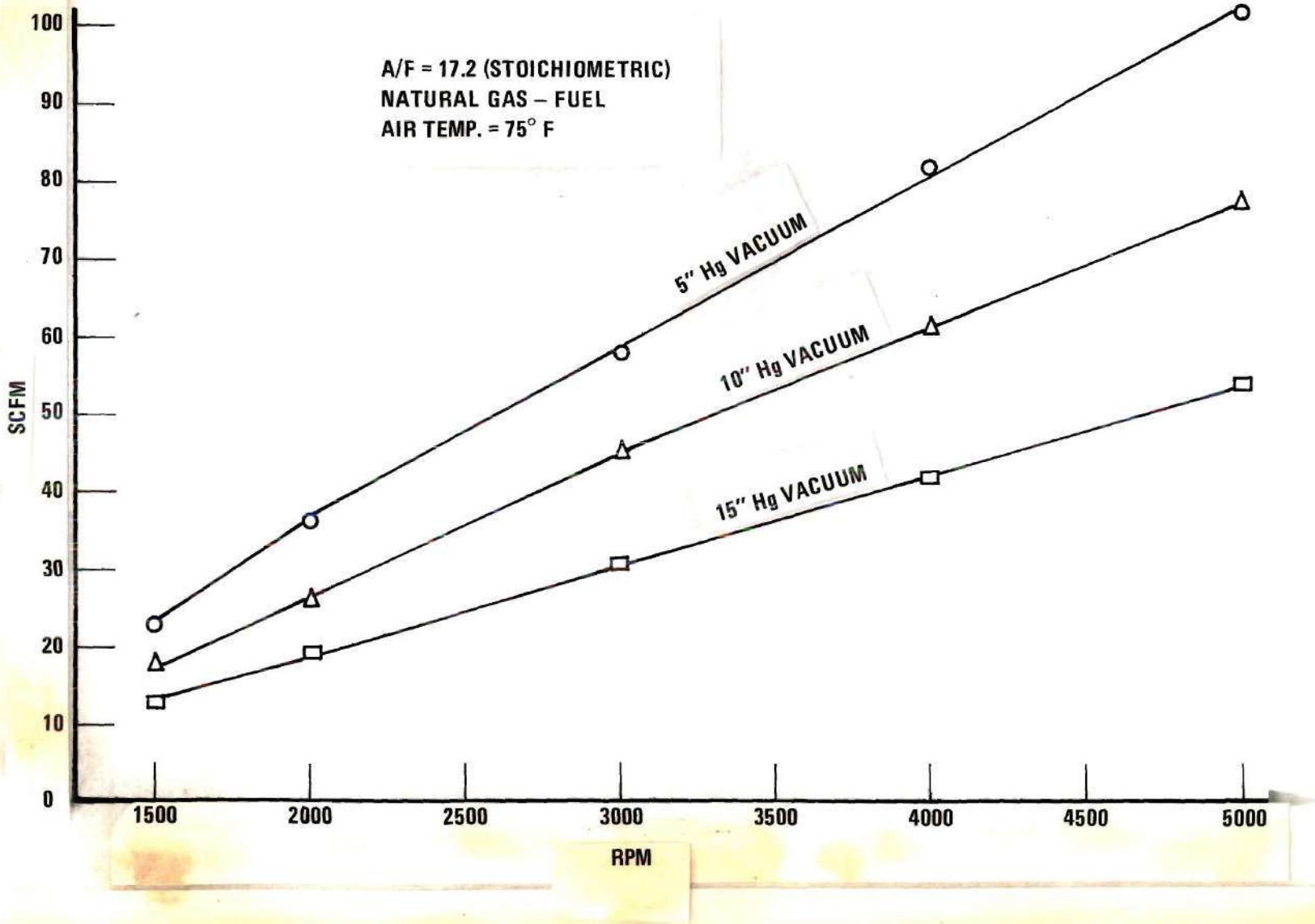

Figure 18. Engine Intake Air versus Engine RPM and Load

APPENDIX F

PERCENTAGE AIR INJECTION VERSUS RPM AND LOAD

Air Injection (Percentage of Engine Intake Air)	RPM	Load (Inches Vacuum)
0.22	1500	5
0.32	1500	10
0.54	1500	15
0.18	2000	5
0.26	2000	10
0.41	2000	15
0.11	2500	5
0.18	2500	10
0.29	2500	15
0.07	3000	5
0.13	3000	10
0.25	3000	15
0.02	4000	5
0.10	4000	10
0.19	4000	15
0.00	5000	5
0.00	5000	10
0.03	5000	15

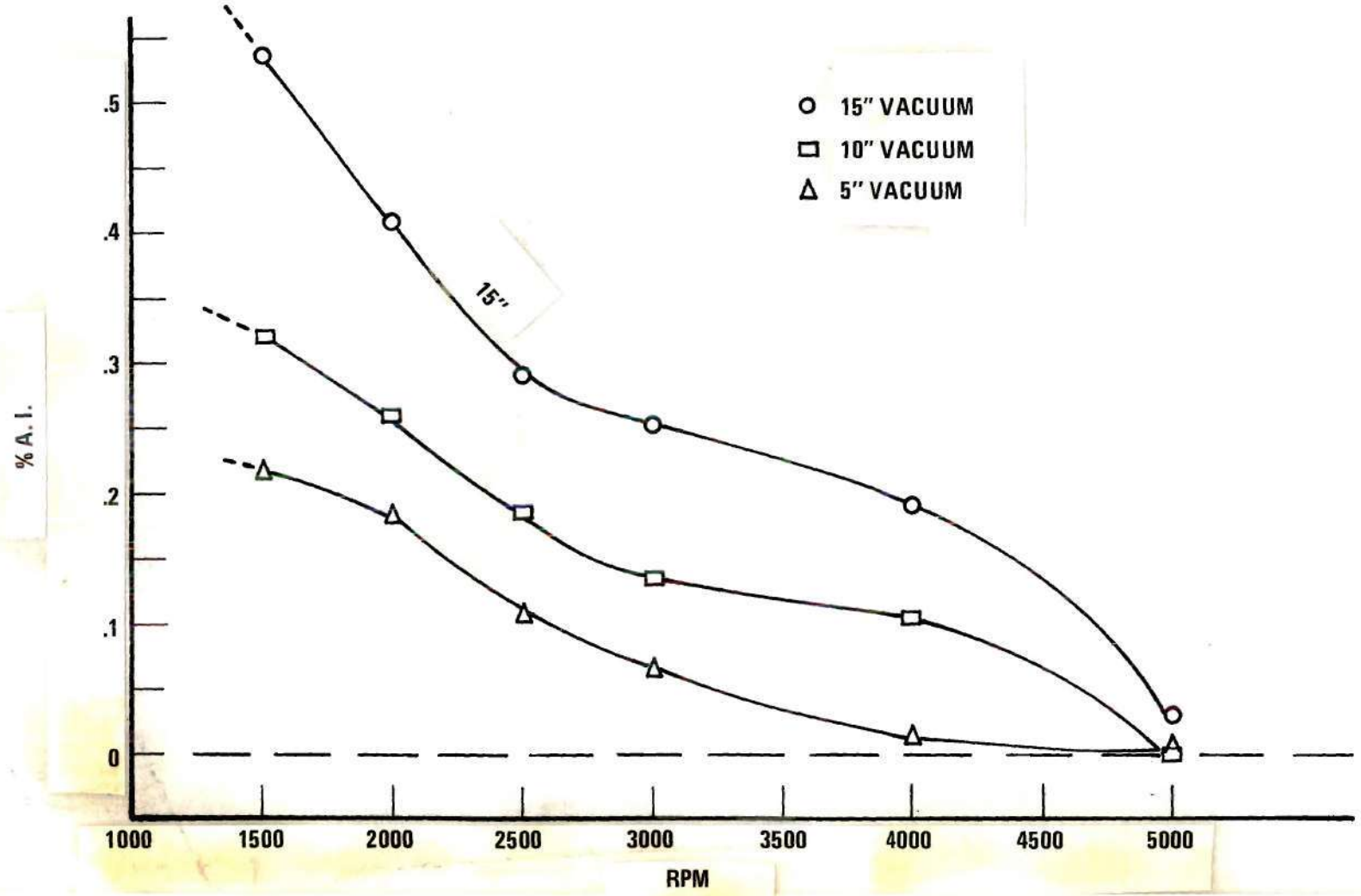

Figure 19. Percentage Air Injection versus Engine RPM and Load

APPENDIX G

Table G1. Emissions Variation with Air Injection [Figure 6]

Mixture: Approximately stoichiometric. Timing: Factory specifications (always).	A.I. (%)	HC (ppm)	CO (%)
1500 rpm 15"Hg Vacuum	0	2780	2.50
	10	2550	2.42
	20	2450	2.32
	30	2420	2.24
	40	2390	2.23
2500 rpm 15"Hg Vacuum	0	1740	1.48
	10	1630	1.44
	20	1570	1.41
	30	1520	1.41
4000 rpm 15"Hg Vacuum	0	1360	1.02
	10	1320	0.98
	20	1290	0.96

Table G2. HC Emissions Variation with Fuel Injection [Figure 7]

Mixture: A/F = 19.0 Full Air Injection	HC (ppm)	Methane Injected (SCFH)
1500 rpm 15"Hg Vacuum Reactor Temperature = 920°F	2600	0
	2800	1
	3040	2
	3800	6
	4000+	10
2500 rpm 10"Hg Vacuum Reactor Temperature = 1380°F	1100	0
	1180	1
	1300	2
	1600	6
	1980	10

Table G3. Emissions Variation with Air-Fuel Mixture

1500 rpm = Engine Speed 15"Hg Vacuum = Engine Load						
A/F	Sampling Point	Air Injection	HC (ppm)	CO (%)	NO (ppm)	Converter Temperature
19.5	1	No	2080	1.05	97	1190°F
	1	Yes	1850	0.90	100	1160
	2	No	1450	0.15	98	1190
	2	Yes	1430	0.05	98	1160
22.0	1	No	1830	0.18	96	960
	1	Yes	1760	0.17	96	960
	2	No	1680	0.02	97	960
	2	Yes	1630	0.02	97	960
17.0	1	No	3350	3.44	95	1380
	1	Yes	3270	3.19	96	1460
	2	No	2830	1.40	96	1380
	2	Yes	2320	0.89	96	1460

Table G4. Emissions Variation with Air-Fuel Mixture

1500 rpm = Engine Speed 10"Hg Vacuum = Engine Load						
A/F	Sampling Point	Air Injection	HC (ppm)	CO (%)	NO (ppm)	Converter Temperature
16.4	1	No	2600	4.70	150	1440°F
	1	Yes	2500	4.60	140	1430
	2	No	2390	3.37	140	1440
	2	Yes	2320	3.24	142	1430
18.6	1	No	1890	1.70	190	1310
	1	Yes	1870	1.65	210	1290
	2	No	1460	0.13	220	1310
	2	Yes	1450	0.11	220	1290
21.4	1	No	1560	0.08	200	1010
	1	Yes	1500	0.09	200	1010
	2	No	1440	0.02	200	1010
	2	Yes	1380	0.02	200	1010

Table G5. Emissions Variations with Air-Fuel Mixture

1500 rpm = Engine Speed 5"Hg Vacuum = Engine Load						
A/F	Sampling Point	Air Injection	HC (ppm)	CO(%)	NO (ppm)	Converter Temperature
19.8	1	No	1760	1.30	430	1330°F
	1	Yes	1700	1.30	430	1480
	2	No	1250	0.05	410	1330
	2	Yes	1210	0.05	410	1480
16.8	1	No	2410	4.70	200	1460
	1	Yes	2350	4.80	200	1500
	2	No	2120	4.69	210	1460
	2	Yes	2050	4.60	210	1500
22.2	1	No	1620	0.09	430	1120
	1	Yes	1550	0.09	430	1110
	2	No	1500	0.02	440	1120
	2	Yes	1490	0.02	420	1110

Table G6. Emissions Variation with Air-Fuel Mixture

2500 rpm = Engine Speed 15"Hg Vacuum = Engine Load						
A/F	Sampling Point	Air Injection	HC (ppm)	CO (%)	NO (ppm)	Converter Temperature
16.0	1	No	2750	3.85	1550	1550°F
	1	Yes	2680	3.68	1350	1550
	2	No	2250	2.10	1300	1550
	2	Yes	2120	1.80	1200	1550
19.8	1	No	1120	0.20	210	1250
	1	Yes	1040	0.19	200	1250
	2	No	960	0.03	200	1250
	2	Yes	920	0.03	200	1250
22.5	1	No	1320	0.10	170	1160
	1	Yes	1290	0.10	170	1160
	2	No	1200	0.02	190	1160
	2	Yes	1150	0.02	190	1160

Table G7. Emissions Variations with Air-Fuel Mixture

2500 rpm = Engine Speed
10"Hg Vacuum = Engine Load

A/F	Sampling Point	Air Injection	HC (ppm)	CO (%)	NO (ppm)	Converter Temperature
15.6	1	No	2300	5.26	180	1590°F
	1	Yes	2050	4.85	130	1590
	2	No	2000	4.95	150	1590
	2	Yes	2020	4.60	150	1590
19.3	1	No	1060	1.05	450	1420
	1	Yes	1060	1.05	480	1420
	2	No	840	0.20	470	1420
	2	Yes	860	0.18	470	1420
21.4	1	No	900	0.10	425	1260
	1	Yes	900	0.10	425	1290
	2	No	850	0.04	405	1260
	2	Yes	850	0.04	405	1290

Table G8. Emissions Variations with Air-Fuel Mixture

2500 rpm = Engine Speed
5"Hg Vacuum = Engine Load

A/F	Sampling Point	Air Injection	HC (ppm)	CO(%)	NO (ppm)	Converter Temperature
15.5	1	No	1350	5.45	510	1600°F
	1	Yes	1300	5.15	310	1580
	2	No	1140	4.90	260	1600
	2	Yes	1160	5.15	270	1580
19.0	1	No	950	0.30	870	1450
	1	Yes	900	0.30	870	1440
	2	No	780	0.05	900	1450
	2	Yes	740	0.05	900	1440
21.7	1	No	820	0.15	690	1330
	1	Yes	750	0.15	550	1330
	2	No	640	0.04	550	1330
	2	Yes	640	0.04	540	1330

Table G9. Emissions Variations with Air-Fuel Mixture

4000 rpm = Engine Speed 15" Hg Vacuum= Engine Load						
A/F	Sampling Point	Air Injection	HC (ppm)	CO (%)	NO (ppm)	Converter Temperature
16.4	1	No	2300	3.30	150	1680°F
	1	Yes	2260	3.25	160	1730
	2	No	2000	1.90	160	1680
	2	Yes	1700	1.95	140	1730
18.8	1	No	870	0.90	190	1550
	1	Yes	910	0.85	200	1550
	2	No	690	0.18	225	1550
	2	Yes	710	0.13	225	1550
21.5	1	No	2000	0.15	170	1330
	1	Yes	1900	0.15	170	1330
	2	No	1710	0.05	170	1330
	2	Yes	1670	0.05	170	1330

Table G10. Emissions Variations with Air-Fuel Mixture

4000 rpm = Engine Speed 10" Hg Vacuum = Engine Load						
A/F	Sampling Point	Air Injection	HC (ppm)	CO (%)	NO (ppm)	Converter Temperature
16.7	1	No	870	3.80	210	1730°F
	1	Yes	870	3.75	220	1740
	2	No	790	3.40	215	1730
	2	Yes	740	3.35	215	1740
19.8	1	No	450	0.60	480	1570
	1	Yes	420	0.50	450	1570
	2	No	230	0.30	440	1570
	2	Yes	200	0.30	440	1570
22.3	1	No	1330	0.20	270	1420
	1	Yes	1290	0.20	280	1420
	2	No	1080	0.02	240	1420
	2	Yes	1220	0.02	240	1420

Table G11. Emissions Variation with Air-Fuel Mixture

4000 rpm = engine speed 5" Hg Vacuum = engine load						
A/F	Sampling Point	Air Injection	HC (ppm)	CO (%)	NO (ppm)	Converter Temperature
16.1	1	No	660	4.50	470	1770°F
	1	Yes	640	4.45	380	1770
	2	No	530	4.55	310	1770
	2	Yes	500	4.50	310	1770
19.7	1	No	130	0.15	630	1680
	1	Yes	110	0.15	680	1680
	2	No	60	0.05	660	1680
	2	Yes	40	0.05	700	1680
21.6	1	No	430	0.25	560	1590
	1	Yes	450	0.25	480	1590
	2	No	310	0.09	350	1590
	2	Yes	290	0.09	350	1590

Table G12. HC Specific Reaction Rate Variation with Temperature and Engine Operating Condition

Engine Operating Condition	Fuel	HC Specific Reaction Rate Numerical Constant	Converter Temperature
1500 rpm 15" Hg Vacuum	Gasoline	1160 $ft^3 lb\text{-}mol^{-1} sec^{-1}$	1300°F
	Propane	240 "	1040
	Natural Gas	70 "	1190
2500 rpm 10" Hg Vacuum	Gasoline	4600 "	1420
	Propane	2400 "	1380
	Natural Gas	320 "	1420
4000 rpm 5" Hg Vacuum	Gasoline	19,600 "	1690
	Propane	11,400 "	1650
	Natural Gas	6000 "	1660

Table G13. Maximum Horsepower Variation with Engine RPM and Fuel (Figure 15)

RPM	Fuel	Road Horsepower
1500	Gasoline	16
	Propane	13
	Natural Gas	10
2000	Gasoline	23
	Propane	21
	Natural Gas	18
2500	Gasoline	30
	Propane	29
	Natural Gas	25
3000	Gasoline	38
	Propane	35
	Natural Gas	29
3500	Gasoline	44
	Propane	41
	Natural Gas	33
4000	Gasoline	50
	Propane	47
	Natural Gas	35
4500	Gasoline	54
	Propane	51
	Natural Gas	36
5000	Gasoline	58
	Propane	54
	Natural Gas	37
5500	Gasoline	62
	Propane	58
	Natural Gas	38
6000	Gasoline	64
	Propane	59
	Natural Gas	39

Table G14. Maximum Torque Variation with Engine Speed and Fuel (Figure 16)

Fuel	RPM	Torque (ft-lbs)
GASOLINE	1500	56.0
	2000	59.0
	2500	63.0
	3000	64.5
	3500	66.5
	4000	65.6
	4500	63.0
	5000	60.5
	5500	59.0
	6000	56.0
PROPANE	1500	45.5
	2000	55.0
	2500	61.0
	3000	61.2
	3500	61.5
	4000	62.0
	4500	59.5
	5000	56.5
	5500	55.5
	6000	51.5
NATURAL GAS	1500	35.0
	2000	47.0
	2500	52.5
	3000	51.0
	3500	49.0
	4000	46.0
	4500	42.0
	5000	39.0
	5500	36.0
	6000	34.5

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